Experimental investigation of surface chemistry on the performance of heat pipes

by

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A thesis submitted in partial fulfillment of the requirements for the degree of

Master of Science

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 $\ensuremath{\mathbb{C}}$ Raihanul Kabir, 2024

Abstract

Much research has been performed over the past few decades on heat pipe performance enhancement. A major thrust of investigation has been to examine factors impacting condensation. Measurements suggest that drop-wise condensation achieves a comparatively larger rate of condensate discharge on smooth and textured surfaces. To promote drop-wise condensation, surface chemical treatments can be used. In some cases (e.g. carbon nanotube coating), the chemical treatments in question increase the overall thermal conductivity. More typically, however, the effect of the chemical treatment is to increase the resistance to heat transfer by conduction. In such instances, there is a trade-off associated with the application of surface chemical treatments because they decrease conduction but facilitate enhanced convection. The current research focuses on the effect of surface energy on the condenser section of a representative flat plate heat pipe. Our experimental design includes a transparent top, and so allows for visual confirmation of the mode of condensation. Observations and measurements can, therefore, be compared against those collected in a small-scale condensation chamber that includes metal coupons having different surface chemical properties.

In this study small coupons of 25.4 mm \times 25.4 mm \times 3 mm (Length \times Breadth \times Thickness) are chemically treated to create hydrophobic and superhydrophobic surfaces and examined in terms of condensation behaviour inside a condensation chamber. Visual data has been analyzed with the help of Image Pro Premier software. In particular, we quantify the degree to which a superhydrophobic coating increases

the degree of water droplet shedding. By extension, we quantify the degree to which applying a low surface energy coating in the condenser section of a heat pipe decreases overall thermal resistance and thereby increases heat pipe performance. Additionally, we also explore the effect of the angle of inclination on the performance of the heat pipe in terms of the temperature profile along the external wall of the heat pipe.

Preface

This thesis is an original work by Raihanul Kabir. The research was funded by NSERC and Engineered Air Ltd. The condensation chamber in chapter 3 and the tilting mechanism of the optical heat pipe in chapter 4 were designed by Ryan Baily (former Lab Manager). The first iteration of the design of the optical heat pipe was completed by Eric Choi (a former undergraduate student) and the final iteration of the CAD design of the optical heat pipe which was used to machine the optical heat pipe parts described in chapter 4 was completed by Lida Su (a former M.Sc. student). A version of Chapters 3 and 4 is in the process of preparation for publication in a peer-reviewed journal. This research has also been presented at the Canadian Chemical Engineering Conference (CSChE 2023) by Raihanul Kabir, Prashant R. Waghmare, and Morris R. Flynn as: "Investigating the effects of surface chemistry on the performance of heat pipes." 'Great things are not done by one person. They are done by a team of people' -Steve Jobs Dedicated to my parents, in laws, my sister and my beloved wife Faria Habib Disha

Acknowledgements

First, I would like to thank the almighty for giving me this opportunity to pursue my long-cherished wish to perform research. I was lucky to have Dr Morris R. Flynn and Dr Prashant R. Waghmare as my supervisors in my Master's program at the University of Alberta. Their support and constructive feedback have helped me a lot during this whole period of completing my Master's. When I look back on the time I started till now, the amount of transformation I had, not only with respect to knowledge but also the number of skills I have developed, all these were somehow inspired by Dr. Morris and Dr. Prashant.

I would like to acknowledge the financial support given by Engineered Air Ltd. and NSERC. I am highly grateful to all the heat pipe project team members. I am grateful to my lab manager Ryan Baily, from whom I got all the safety and onboarding training and got trained on operating lab equipments. I would like to thank my colleagues Vivek Kumar, Lida Su, Abrar Ahmed, Ganesh Prabhu Komaragiri, and Zachary Shannon, for all the constant assistance and support throughout the past few years.

Finally, I would like to thank my mother, Jahanara Begum, and my father, S M Kabir, for always inspiring me. Thanks to my sister, Afia Fahmida Dona, for supporting my parents back home in Bangladesh while I stayed away in Canada. Thanks to my in-laws for all the love and support. Last but not least, I would like to thank my dear wife, Faria Habib Disha, for supporting me financially and emotionally while I keep working to fulfill my dreams.

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List of Symbols

Latin

 θ Water surface contact angle

Greek

 α Angle of inclination

R Overall thermal resistance

 $^{\circ}\mathrm{C/W}$

Chapter 1 Introduction

A heat pipe is a passive device that efficiently transfers heat from a hot surface to a cold one through phase change, involving boiling and condensation of a working fluid. Renowned for their superior heat transfer capabilities, heat pipes operate by absorbing heat as the fluid boils and releasing it upon condensation. Their exceptional performance makes them indispensable in various engineering applications, outperforming other heat transfer systems even when positioned vertically or upside down [1]. By leveraging phase change, heat pipes are associated with a high thermal conductivity, facilitating rapid heat transfer with minimal temperature differentials. In electronics, such as laptops and desktops, heat pipes efficiently cool chips and circuits, which is crucial given the ongoing challenge of component overheating due to miniaturization [2]. In aerospace applications, heat pipes precisely regulate temperatures, even in the vacuum of space where gravity is absent, highlighting their suitability for extraterrestrial use [3]. Moreover, they play a vital role in renewable energy by minimizing heat loss during solar energy collection, ensuring maximum power generation or heating performance. Medical equipment cooling also benefits from heat pipes, where precise temperature control is essential.

1.1 Overview of heat pipe technology

Richard S. Gauglar, an engineer at the General Motors Corporation, introduced the concept of heat pipes in 1942, although it was George M. Grover who coined the term "heat pipe" in 1963 [4]. Grover independently developed a similar device, which he later patented, earning him recognition as the inventor of heat pipes. Since the 1960s, extensive research has been dedicated to advancing heat pipe technology, leading to the creation of various types of heat pipes.



Figure 1.1: Working principle of a conventional heat pipe : Red arrows represent heat input and the indigo color arrows represent heat release, the direction of vapor flow is shown using white arrows, and the direction of liquid flow through the wick is represented by blue arrows.

1.1.1 Heat pipe structure

Irrespective of the various types, a heat pipe typically consists of the following components:

- Evaporator section: The region that is in thermal contact with the hotter surface is termed as the evaporator section. As the name suggests, evaporation is the heat transfer process in this section.
- Adiabatic section: Vapor travels to the colder region through this section. Ideally, there is no heat transfer associated with this section. In some designs, heat pipes do not even have an adiabatic section.

- Condenser section: The region that is in thermal contact with the colder surface and responsible for the condensation process is termed as the condenser section. Condensate forms in this section and returns to the evaporator section either by assistance of gravity in case of gravity-assisted heat pipes or by capillary action in case of wicked heat pipes.
- Wick structure: A wick is an inner lining surface that facilitates capillary action and assists in the liquid or condensate return to the evaporator, which is responsible for the continuous functioning of a heat pipe. Wicks might not even be present in some heat pipes, in which case the heat pipe is called a wickless heat pipe or thermosyphon [5].
- Working fluid: The fluid that acts as the heat transfer medium inside of the heat pipe is called the working fluid. The selection of this working fluid depends on the application of the heat pipe and the temperature range in which it has to operate.
- Container: Typically copper or aluminum is used as the container or shell material because of their higher thermal conductivities. The conventional heat pipes include cylindrical, flat plate, and rotating heat pipes [6]. (A brief overview of conventional heat pipes is given in chapter 2.) The container houses the working fluid of the heat pipe.

Figure 1.1 illustrates a schematic of a conventional cylindrical heat pipe.

1.1.2 Heat pipe working principle

The working principle of a heat pipe can be explained in two stages. They are as follows:

• Evaporation: Heat is absorbed by the working fluid from the evaporator section. After heat absorption, the working fluid undergoes a phase change from liquid to vapor by evaporation. The vapor then travels through the heat pipe and reaches the condenser region. The driving force for vapor flow is the vapor pressure difference between the condenser and evaporator section, which is created due to the formation of vapor in the evaporator section and the condensation of vapor in the condenser section.

• Condensation: At the condenser section, vapor releases heat as it condenses, and the working fluid changes its phase from vapor to liquid. The condensate is returned to the evaporator section by either a capillary mechanism in the case of a wicked heat pipe or with the help of gravity in the case of a gravity-assisted heat pipe [7].

Figure 1.1 illustrates the above-mentioned two stages of an operating heat pipe.

1.1.3 Heat pipe application

Heat pipes are applied in various industries owing to their versatile thermal management capabilities. A short overview of some notable applications of heat pipes is discussed below.

Electronics industry

Heat pipes are used in electronic devices such as laptops to dissipate heat. The central processing unit (CPU) and graphics processing unit (GPU) produce a significant amount of heat during operation for which heat pipes are necessary to maintain optimal performance and prevent thermal throttling. In ultrabooks and slim laptops, where space is a major concern, flat and compact heat pipes are often integrated to efficiently dissipate heat without adding extra mass. Heat pipes' slim design and high thermal conductivity make them ideal for thermal management in thin, lightweight devices. As a result, heat pipes are integrated into laptop cooling systems alongside fans and heat sinks to provide comprehensive thermal management solutions. Likewise, heat pipes are also integrated into smartphones and tablets where space is a concern. Micro heat pipes and vapor chamber heat pipes are commonly used due to their miniature size and high thermal conductivity [8]. Pulsating heat pipes (PHP) are also used in confined and compact spaces to transport heat over short distances. These compact heat pipes enable efficient heat dissipation without compromising device design or performance. In LED lighting fixtures, heat pipes effectively dissipate heat generated by LEDs, which enhances their longevity and sustains consistent performance levels.

Servers and data centers have many high-performance computing components in a confined space, where heat generation is a big challenge. Heat pipes and other cooling technologies, like liquid cooling and air conditioning, are employed to manage heat transfer [9].

Automotive industry

Heat pipes are essential for cooling electric vehicle (EV) batteries, which enhances their performance and extends their lifespan by efficiently drawing heat away from the battery cells [10]. The battery efficiency is increased by maintaining the battery at its optimal operating temperature [11]. Sintered powder heat pipes are used in automotive engine cooling systems to manage the engine's heat, thereby improving fuel efficiency. These types of heat pipes offer compact design and high thermal conductivity, which helps in efficient heat transfer in automotive applications. Heat pipes help maintain the engine's thermal stability and prevent overheating, reducing the risk of engine damage and costly repairs.

Efficient engine cooling by heat pipes also contributes to reducing harmful emissions, such as nitrogen oxides (NO_x) and hydrocarbons (HC) [12]. Heat pipes optimize combustion efficiency by maintaining optimal operating temperatures, ultimately leading to cleaner exhaust emissions and greater compliance with emission regulations. Some vehicles have turbochargers, which generate significant heat due to air compression in the intake phase of engine combustion. Heat pipes are applied to efficiently dissipate this heat, preventing overheating the turbocharger housing and associated components [13]. This helps to improve engine power output and responsiveness, particularly in high-performance vehicles.

Heat pipes are also used in the transmission cooling systems to regulate transmission fluid temperatures and prevent overheating. The transmission fluid heat pipes help maintain optimal viscosity and lubrication properties, ensuring smooth gear shifting and prolonged transmission life.

Aerospace industry

Heat pipes play a crucial role in spacecraft thermal management by maintaining temperatures within specified ranges. Onboard equipment such as electronics and propulsion systems generate heat during operation. Maintaining stable temperatures ensures these components function properly in extreme space environments. Moreover, heat pipes are reliable in spacecraft thermal control systems as they do not require moving parts or any external power source. Heat pipes used in spacecraft are made from lightweight and durable materials such as aluminum, titanium, or composite materials. Their design is optimized to withstand launch vibrations, microgravity conditions, and thermal cycling experienced during space missions.

Heat pipes are included in the thermal protection systems of re-entry vehicles, such as space capsules and atmospheric entry probes, to dissipate the intense heat generated during atmospheric re-entry. They help protect the vehicle and its occupants or payload from high temperatures and thermal stresses during re-entry. Sintered powder heat pipes are used as cooling systems in aircraft electronics compartments to ensure reliable operation. Aircraft engines also use heat pipes to manage heat generated by combustion processes and engine components. They help dissipate excess heat from critical engine parts, such as turbine blades and combustion chambers, improving engine efficiency and reliability [14].

Communication systems and electronic components generate heat during the operation of satellites. Heat pipes are used to efficiently dissipate this heat and maintain optimal operating temperatures, ensuring the reliability and longevity of satellite systems. They facilitate thermal regulation in orbit by transferring excess heat from sun-facing components. This helps prevent overheating of critical systems and ensures continuous operation of satellites throughout their mission lifespan. Capillary pumped loop heat pipes (CPL) are used in satellite thermal management systems to maintain stable temperatures for onboard electronics and instruments [15]. They utilize capillary action to circulate working fluid, enabling precise temperature control in microgravity environments.

Renewable energy

Heat pipes are vital in solar thermal systems. Aluminum heat pipes are found to be effective for application in solar collectors which improves the performance of solar thermal systems by maximizing energy capture and conversion [16]. Heat pipes facilitate transferring excess heat to thermal storage tanks or materials during periods of high solar radiation. This stored thermal energy can be utilized for heating or electricity generation during periods of low sunlight or at night, improving system reliability and energy availability.

In geothermal energy applications, heat pipes are utilized in power plants to transport heat from underground heat sources, such as hot rock formations or geothermal reservoirs, to the surface, where it can be converted into electricity [17]. Their ability to transfer heat over long distances with minimal energy loss makes them well-suited for geothermal applications.

Heat pipes can also be included in biomass gasification systems to transfer heat to biomass feedstocks. This helps in thermochemical conversion processes such as pyrolysis and gasification [18]. Heat pipes also control the temperature in biomass reactors, improving gasification performance and achieving higher energy yields.

In combined heat and power systems (CHP), heat pipes are employed for waste heat recovery from power generation processes (e.g., gas turbines, engines, etc) [19, 20]. Heat pipes enable efficient heat transfer to heat recovery steam generators or district heating networks, which maximizes the overall energy efficiency and sustainability of combined heat and power installations.

HVAC

In HVAC systems, heat pipes transfer heat from exhaust air to incoming fresh air, thereby boosting energy efficiency and alleviating heating and cooling burdens on the system. During the winter, heat from the warm exhaust air is transferred to the colder fresh air, preheating the cold air before entering the building. Conversely, heat from the hot fresh air is transferred to the cooler exhaust air, precooling the hot air during the summer. This process reduces HVAC heating and cooling loads leading to significant energy savings and improved efficiency. By recovering heat from exhaust air, heat recovery systems help maintain indoor air quality by reducing the need for constant ventilation and minimizing energy losses associated with heating or cooling incoming fresh air. This ensures a constant supply of fresh air while mitigating energy consumption and operating costs.

Copper heat pipes efficiently transfer heat from the evaporator coils in refrigeration systems, improving cooling efficiency and reducing energy consumption. As refrigerant vapor flows through the evaporator coils, heat pipes absorb heat from the surrounding air or products, effectively cooling the space or refrigerated goods. Heat pipes assist in dehumidification in air conditioning and refrigeration applications by removing moisture from the air [21].

As warm, humid air encounters the cool surface of the heat pipe, water vapor condenses, and the condensed liquid is collected, reducing indoor humidity levels and improving thermal comfort.



Figure 1.2: Notable applications of heat pipes

1.2 Research gap and research motivation

The parameters that quantify the performance of heat pipes are overall thermal resistance and thermal conductivity, which have an inverse relationship. Both variables are influenced by many factors. For example, the properties of the working fluid can greatly influence heat pipe performance. Some other factors include fill fraction of the working fluid, heat input, and inclination angle. Much research has been performed to quantify the overall thermal resistance and its dependence on the aforementioned factors. For experimental studies, the temperature difference between the evaporator and condenser is measured to calculate the overall thermal resistance using

$$R = \frac{T_e - T_c}{Q} \tag{1.1}$$

In this equation, R represents the overall thermal resistance, T_e represents evaporator temperature, T_c represents condenser temperature, and Q represents the heat input to the heat pipe system. The lower the value of the overall thermal resistance, the better the heat transfer performance from hot surface to cold surface. Presently, because of the usage of powerful and compact computing devices, a large amount of heat is generated, which reduces the longevity of these devices. The present heat pipe technology is not enough to manage the heat load [22]. As a result, improving heat pipe performance is crucial for addressing thermal engineering challenges across various industries, which were discussed in subsection 1.1.3. The significance of this endeavor can be summarized as follows:

- Due to the miniaturization of appliances, space and weight are significant constraints. As a result, designing new, efficient, and smaller heat pipes is important [23].
- With the current trend toward miniaturization of electronic chips, effectively dissipating the heat produced on these chips is critical to ensure the longevity of electronic devices. By moving heat across potentially nontrivial distances, heat pipes have an important role in this process of heat dissipation [2].
- Heat energy accounts for a large portion of global energy consumption [24]. Due to the rapid increase in worldwide energy demand, enhancement of the heat transfer performance of heat pipes is essential to minimize energy loss in the form of heat, especially in renewable energy systems like solar thermal systems, geothermal systems, and combined heat and power systems.

Researchers have been studying different techniques to increase heat pipe performance for the past couple of decades. Some notable approaches taken by past researchers are discussed below:

1.2.1 Wick structure optimization

The literature is replete with various examples of wick designs, which may differ in wick porosity, groove structure, and composite wick structure. Studies which are related to the optimization of wick structures try to increase capillary pumping by wick structure modification. In a relevant experimental study by Kumaresan et al. [25], the effect of wick structure was investigated by comparing the performance of wicks classified as mesh-type, sintered, and composite. The results demonstrated a maximum thermal resistance reduction of 34.65% for the composite wick vs. the sintered wick and of 42.28% for the composite wick vs. the mesh wick. This is because the wick porosity and wick permeability of the composite wick structure make it easier to return working fluid from the condenser section. In another experimental study done by G. Franchi and X. Huang [26], different wicks were compared to investigate the effect of wick structure on the overall heat pipe thermal conductivity. They fabricated wicks from a biporous structure from nickel powders sintered onto a layer of coarse pore copper mesh. The researchers of this particular study concluded that with the presence of both fine and coarse pores, the proposed biporous wicks increased the capillary pumping capability and also maintained a high liquid permeability. An ideal wick structure should have coarse and fine pores which, enhance the capillary pumping, resulting and maintaining high liquid permeability, which increases the condensate return rate to the evaporator of a heat pipe and so reduces the chance of dry out in the heat pipe.

1.2.2 Selection of working fluid

Working fluid is an important component of a heat pipe. Different working fluids possess different thermophysical properties, affecting heat pipe performance [27, 28]. The properties of an ideal working fluid should include the following:

- High thermal conductivity to transfer heat along the length of the heat pipe efficiently. This ensures rapid heat dissipation and effective thermal management.
- A low vaporization temperature allows the working fluid to vaporize quickly at relatively low temperatures.

- A high latent heat of vaporization enables the working fluid to absorb large amounts of heat during vaporization without a significant increase in temperature. This enhances the heat transfer capacity and heat pipe performance [29].
- Chemical stability over a wide range of temperatures prevents degradation or corrosion of the heat pipe materials. Chemical stability thereby ensures longterm reliability and durability of the heat pipe system.
- A low viscosity facilitates fluid circulation.By contrast, high fluid viscosity, particularly in the liquid phase, can impede the working fluid flow within the heat pipe, which reduces heat transfer performance and increases pressure drop.
- A low boiling point is desirable for efficient heat transfer; however, extremely low boiling points may lead to premature vaporization, limiting the heat pipe's performance.

Some commonly used working fluids are water, ethanol, methanol, ammonia, R134a, R513a, etc. [30–32]. A relevant study on applying a phase change material (PCM) as a working fluid by Zhao et al. [33] saw the addition of paraffin wax to a closed-loop oscillating heat pipe (CLOHP) under different heat inputs from 10 W to 80 W. It was concluded that the paraffin wax could not melt completely due to reduced temperature in the case of PCM coupled with CLOHP as a result of the strong thermal conductivity of the CLOHP.

The thermal resistance of heat pipes depends on the heat pipe's fill fraction. A heat pipe with a lower working fluid ratio can decrease heat transfer performance as the evaporator has less working fluid to transfer heat. On the other hand, overfilling the heat pipe can also lead to worse heat transfer performance. As a result, researchers focused on finding the appropriate working fluid ratio that can give the least thermal resistance possible for the given conditions of heat input and heat pipe design parameters. In an experimental study done by T. Sukhchana and C. Jaiboonma [34], the effect of the fill fraction of R-134a working fluid was investigated. They filled a copper heat pipe with 10, 15, and 20 % of working fluid and tested it for 1.97 to 9.87 kW/m^2 heat inputs. They concluded the optimum fill fraction to be 15 %. Note, however, that the optimum fill fraction will assume different values for different working fluids. In a study by Barua et al. [35], the comparison between ethanol and water as working fluids was done based on the working fluid ratio. The experimental results concluded that in the case of water, the thermal resistance for both high and low heat input was minimal at a lower fill fraction of 30 %. But for ethanol at low heat input, the optimum performance was obtained at a higher fill fractions in the case of high heat input. The authors, therefore, concluded that ethanol is a better-working fluid than water for high-heat input applications.

As an endeavor to select a working fluid, another approach past researchers have taken is the integration of nanoparticles with a working fluid, which is why they are called nanofluids. Integration of nanoparticles with working fluid impacts heat pipe performance to a greater extent. Presently a lot of research work is focused on integration of nanofluids to increase the thermal conductivity and heat transfer coefficient of the base working fluid [36–38]. Since this research does not focus on studies related to nanofluid integration, it is not discussed in detail in this thesis.

1.2.3 Surface modification

One of the approaches past researchers have taken to study how to increase the performance of heat pipes is the modification of the internal surface of heat pipes, to promote a more desirable condensation regime. Because this topic is central to the objectives of the present thesis, we revisit this subject just after introducing figure 1.3 below.

Complementing the texts in sections 1.2.1, 1.2.2 and 1.2.3, an overview of the approaches taken by past researchers to enhance the performance of heat pipes is given in figure 1.3.



Figure 1.3: Some approaches taken by past researchers to enhance heat pipe performance

Promotion of liquid return to improve heat pipe performance

One of the techniques followed by past researchers to modify heat pipe performance is the promotion of liquid return of condensate to the evaporator of a heat pipe [39, 40]. The liquid return flow rate depends on the mode of condensation, which is influenced by the wettability of the condensing surface. There are two types of condensation, namely filmwise condensation and dropwise condensation. When a condensing surface exhibits a hydrophilic wettability regime, the condensation mode is film-wise condensation, and hydrophobic wettability exhibits dropwise condensation. In film-wise condensation, a film of condensate forms on the condensing surface, which gradually grows in thickness as the condensation process continues. In the case of dropwise condensation, the condensation process consists of the formation, growth, coalescence and shedding of droplets. There are two popular hypotheses that explain the droplet formation, namely the film rupture hypothesis and the nucleation hypothesis. According to the former [41], a thin layer of condensate initially forms on the surface, which, after reaching a critical thickness, breaks up into droplets due to surface tension. According to the latter hypothesis [42], tiny droplets form at specific nucleation sites and grow by coalescing with neighbouring droplets or by direct condensation until the droplets reach a critical size. Thereafter, body forces cause the droplet to be shed from the surface. As these droplets shed, they clear the surface, allowing new droplets to form in their wake through a process called renucleation. The question of whether dropwise or film-wise condensation dominates depends on a variety of factors e.g. the microstructural details of the surface. Also important is the degree of subcooling, which is the difference in temperature between the condensing surface and the vapor to be condensed [43]. When the subcooling temperature is small and the surface contact angle is large such that the surface is hydrophobic or superhydrophobic, the mechanism of condensation is by nucleation. On the other hand, if the subcooling temperature is large, droplets appear via film rupture [43].

In 1920, Schmidt et al. [44] performed an experimental study that identified dropwise condensation as having five to seven times greater heat transfer coefficients vs. filmwise condensation. The difference arises because in dropwise condensation, droplet roll-off action results in the departure of droplets from the condensing surface after reaching a critical mass which in turn helps to expose the condensing surface again for repeated nucleation. However, in the case of film-wise condensation, because of the film of condensed liquid present in the condensing surface, there is always additional convective resistance in the heat transfer process. As a result, the heat transfer rate is reduced. A relevant study by Rainieri and the research team [39], in 2008 showed an increase of convective heat transfer by 25% when compared between dropwise and filmwise condensation. Another study by Zhao et al.[40] suggests that a hydrophobic condenser section improves the overall heat pipe performance. Although the results of this study suggest good evidence for applying a hydrophobic surface, there was no visual access inside the heat pipe.

Chemical treatment can be applied to modify the heat pipe (interior) surface, which can increase the hydrophobicity and thus promote dropwise condensation. However, doing so in some cases, for example, carbon nanotube coating on a solid surface, can decrease the overall thermal conductivity of the composite solid surface consisting of substrate and coating. As such, a trade-off is associated with applying surface chemical coatings that generally favor hydrophobic layers that are thick but not too thick. We shall revisit the details of this trade-off in quantitative detail in chapter 3 of this thesis.

Notwithstanding the importance of understanding and categorizing the qualitative nature of the condensation process, to the best of our knowledge, only a few experimental studies (we mentioned the related studies in chapter 2 of this thesis) explored the option of optical access to the condenser section of a heat pipe. Visualization experiments can give important insights into the dynamics of droplets formed on the condenser region of the heat pipe. Challenges such as condensation occurring on the heat pipe visual window and thereby obscuring the view make it difficult to study the behavior of condensation phenomena inside of a heat pipe. These challenges can be overcome if the heat pipe is insulated and the insulation is only removed periodically when collecting snapshot images. This strategy has been reported in the Zhao et al.s' [40] research work.

1.3 Objectives

The motivation and objective of this particular study is to investigate the optimal surface energy to modify heat transfer performance, which is related to condensation behavior and droplet shedding performance. We also aim to have optical access inside the heat pipe. In a complementary effort, the aim is to explore the change in the performance of a heat pipe due to different surface energy on the condenser section of a representative flat plate heat pipe. Sample substrates will be prepared to investigate the condensation performance in terms of liquid shedding and then to apply the most promising surface chemical coating inside the condenser section of a heat pipe. The goal is to investigate the effect of surface energy on the performance of a heat pipe. Figure 1.4 shows the steps to achieve the above-mentioned objective. The research activities are divided into two parts:

- Investigation of the role of surface energy on droplet formation and shedding within a condensation chamber experiment.
- Investigation of the role of surface energy within a heat pipe experiment.

A condensation chamber was designed to investigate the droplet shedding performance of the prepared samples with different surface energy and visualize the condensation behavior of these samples. After obtaining quantified and visual information on which surface coating showed the best droplet shedding performance in the small-scale condensation chamber experiment, the optimal surface chemical coating was applied to the condenser section of a purpose-built heat pipe. We named our heat pipe apparatus the 'optical heat pipe,' which, as the name suggests, allows us to see the inside of the heat pipe and thereby characterize the condensation mode.

1.4 Outline

This chapter discusses the research gap and objectives, providing an overview of the steps taken to achieve these objectives.

Chapter 2 contains the literature review on heat pipes involving surface modification and visualization experiments.



Figure 1.4: Experimental flow diagram

Chapter 3 discusses the sample preparation of the substrates for the condenser chamber experiment. It also provides a detailed overview of the condensation chamber, including the experimental steps followed. Lastly, the footage taken during the experiments were analyzed to gather quantified information. This chapter also discusses the outcomes of the condensation chamber experiments.

Chapter 4 discusses the optical heat pipe experiments. The experimental design of the optical heat pipe is also given in this chapter, followed by the experimental steps used during the experiments. The results of the heat pipe experiments are also discussed.

The final chapter discusses the overall conclusion of all the experiments performed in this research and recommendations for future studies.

Chapter 2 Literature Review

2.1 Abstract

In Chapter 1, we listed the factors that affect the performance of heat pipes. Numerous studies have explored different aspects of heat pipes, and the studies in question have deployed theoretical modeling, numerical simulations, and experimental investigations to characterize factors affecting heat transfer behavior [45–48]. After identifying these factors, research has focused on improving heat transfer performance by optimizing wick structures, surface modifications, integrating nanofluids, etc. This chapter delves into historical developments and the classification of heat pipes. In addition, this chapter also discusses recent studies exploring the applicability of chemically modified surfaces to increase heat transfer.

2.2 Introduction

It is important to understand the historical development of heat pipes to understand the objectives of this research. A brief literature review discusses the early developments of heat pipes, the various types of heat pipes, and recent studies relating to surface chemically modified heat pipes.

2.3 Historical development of heat pipe

The origin of the heat pipe concept can be traced back to the Perkins tube, which Jacob Perkins pioneered in the 19th to 20th century. Subsequent refinement was done by Angier March Perkins and his son Ludlow Patton Perkins. Ludlow is officially credited with the Perkins tube patent [49]. The Perkins tube is a wickless gravityassisted closed tube in which water serves as the working fluid and operates in a two-phase cycle. Later, in 1942, Gaugler of the General Motors Corp, who was working on a refrigeration problem, introduced a capillary-based concept of the heat pipe, eliminating the need for gravitational assistance [2]. Gaugler patented the concept in 1944 [50]. Although Gaugler patented the concept, since the necessity of such a device was not clear at that time, no further development was completed and the idea lay dormant until 1962, when Trefethen resurrected the idea of heat pipe as a suggestion in connection with space applications. In 1963, George Grover and his teammates independently invented heat pipe, and tremendous development was completed on several iterations of the heat pipe; Grover also coined the term "heat pipe" and earned recognition as the inventor of the heat pipe.

As mentioned in the above paragraph, the initial utilization and evolution of heat pipes predominantly occurred within the space industry because of the effectiveness of heat pipes in transferring significant heat under reduced gravity conditions facilitated by capillary action [50]. However, realizing the high energy costs in industrial sectors (and therefore the potential savings that heat pipes could afford) quickly redefined the potential of heat pipes for widespread industrial applications and energy conservation [2].

2.4 Heat pipe classification

Based on the design or geometry of conventional heat pipes, these passive heat transfer devices can be divided into five types: cylindrical, rotating, loop, variable conductance, and pulsating [51]. Heat pipes can also be divided into two types based on the presence of wick structures. They are wickless heat pipes, commonly known as thermosyphons, and capillary-driven heat pipes. Below is a brief description of each type of heat pipe, along with some recent studies conducted.

Cylindrical heat pipe

Cylindrical heat pipes, as their name implies, feature a cylindrical design. This kind of heat pipe can either be wicked or wickless. If the heat pipe has a wick structure inside the cylindrical-shaped tube, it is a capillary-driven cylindrical heat pipe. The wick helps capillary action, which is crucial in the heat transfer performance of the heat pipe. A recent numerical analysis conducted by Dev Roy et al. [52] highlighted the significance of a novel stepped corrugated wick geometry to reduce thermal resistance. In this study, which is performed using COMSOL Multiphysics software, the authors revealed a remarkable 75% reduction in thermal resistance compared to smooth wickless heat pipes, underscoring the efficacy of the proposed wick design. Wickless heat pipes are known as thermosyphons, which are gravity-assisted heat pipes. In this kind of heat pipe, the evaporator is at the bottom, and the condenser section is at the top. Liquid returns to the evaporator after condensation with the help of gravity. Moreover, the performance of cylindrical heat pipes is influenced by various factors including working fluid fill fraction, inclination angle, and heat input. Mahdavi et al. [53] investigated the impact of these parameters experimentally, demonstrating, for example, that maintaining the evaporator below the condenser leads to superior heat transfer performance compared to the reverse orientation. It is also observed that underfilling or overfilling the heat pipe impacts the overall thermal
resistance because underfilling the heat pipe reduces the capillary limit, and overfilling the heat pipe interferes with the evaporation and condensation cycle. Experimental results were used to validate a previously developed numerical model. The comparison between numerically predicted results and experimental data showed very good agreement, validating the accuracy of the numerical technique previously reported by the same authors [54]. A schematic diagram of a cylindrical heat pipe with its different sections is shown in chapter 1 in figure 1.1.

Rotating heat pipe

A rotating heat pipe consists of a cylindrical shaft containing a working fluid. The pressure difference propels the vapor between the evaporator and condenser. Rotation of the heat pipe generates a centrifugal force that induces a hydrostatic pressure gradient, which aids in the liquid return process to the evaporator. Vernon H. Gray first proposed the concept [55]. Gray demonstrated the potentiality of a rotating heat pipe comparing it with a similar stationary heat pipe. The working principle of the rotating cylindrical heat pipe is the same as that of a cylindrical heat pipe. However, internal tapers are applied to return the condensate to the evaporator by centrifugal force because capillary wicks are not used inside the rotating heat pipe. Rotating heat pipes cool the rotating parts of motors and powered machine tools such as drill bits and milling machines. Figure shows a schematic of a cylindrical-shaped rotating heat pipe.

Loop heat pipe

A loop heat pipe operates on the same principles of evaporation and condensation as conventional heat pipes. It was introduced in 1971 by Gerasimov and Maydanik [57]. Loop heat pipes consist of a capillary pump (or evaporator), a compensation chamber (or reservoir), a condenser, and liquid and vapor lines. Wicks are present in the evaporator and compensation chamber. The latter handles excess liquid within



Figure 2.1: Schematic diagram of a rotating heat pipe, here ω represents angular velocity and r represents radius of heat pipe (From [56]; used with permission.)

the loop heat pipe during regular operation [58]. A secondary wick with larger pores is sometimes used in the design which links the evaporator to the compensation chamber to supply the primary wick with liquid, particularly when the compensation chamber is positioned below the evaporator or when the loop heat pipe operates in microgravity conditions. Loop heat pipes are used in space and electronic industries.



Figure 2.2: Schematic diagram of a loop heat pipe (From [57]; used with permission.)

Variable conductance heat pipe

A variable conductance heat pipe is similar to a capillary-driven or two-phase thermosyphon. It includes non-condensing gas in vapor space. Due to the absence of condensation where the non-condensing gas exists, a condenser section is rendered partially which is ineffective in transferring heat to the heat sink. When the heat input to the evaporator section rises, both vapor and gas temperatures increase, causing compression of the inert gas. Consequently, more condenser surface area becomes accessible for heat transfer [24]. Figure 2.3 shows the schematic diagram of a variable conductance heat pipe. This kind of heat pipe is applicable where the heat input is variable. They are mostly used in electronics and vehicles. V. Kravets et al. [59] present a heat pipe controlling device demonstrated on the electronic unit of DLR's MASCOT lander deployed on the NEA sample return mission, Hayabusa 2 (JAXA) which shows the potentiality of using variable conductance heat pipes in the space industry. The focus was developing and testing heat pipes with variable thermal conductance within a specific temperature range. Significant improvement in thermal conductivity, ranging from 0.04 W/K to 2.1 W/K, was observed across temperatures from -20° C to $+55^{\circ}$ C.



Figure 2.3: Schematic diagram of a variable conductance heat pipe (From [60]; used with permission.)

Pulsating heat pipe

The pulsating heat pipe consists of a lengthy capillary tube bent multiple times, positioning the evaporator and condenser sections at these bends. Based on whether the ends of the heat pipe are connected or not, there are two types: the looped pulsating heat pipe and the unlooped pulsating heat pipe. Due to the counterflow of liquid and vapor in any conventional heat pipe, there is an entrainment limit present in other types of heat pipe. But in pulsating heat pipes there is no counterflow and therefore no limitation imposed by entrainment. These heat pipes are typically used in electronic industries. Figure 2.4 shows a schematic diagram of a looped and unlooped pulsating heat pipe.



Figure 2.4: Schematic diagram of pulsating heat pipe: looped pulsating heat pipe on the left and unlooped pulsating heat pipe on the right (From [61]; used with permission.)

2.5 Factors that influence the performance of heat pipe

Several factors affect the performance of heat transfer. A brief discussion of some of the notable factors which affect heat transfer performance is given below:

2.5.1 Wick structure

A wick is a porous structure made of steel, aluminum, copper, carbon fiber, etc [29]. Wicks can vary on the size of pore. The purpose of the wick is to transport liquid from the condenser to the evaporator using capillary pressure. There are different types of wick structures. They are:

- Sintered powder metal
- Grooved wick
- Wire screen mesh
- Fiber wick

Some heat pipe applications require a wick structure to return liquid to the evaporator, where gravity is unavailable. The type of wick chosen for a particular design is often dictated by the physical properties of the working fluid. For example, working fluid viscosity, surface tension, etc. The permeability and porosity of the wick structure are considered for wick selection because they influence liquid return from the condenser to the evaporator, which affects the overall thermal resistance. Each wick structure has some advantages and disadvantages. A brief description of each type of wick structure is given below.

Sintered powder metal wick

Sintered powder metal wicks are typically composed of porous metals. Common materials used in the composition of sintered powder metal wicks are stainless steel, copper, aluminum, and their alloys. Advantages of sintered powder metal wicks include the following:

• There is a trade-off between high and low porosity wicks. Excessively high porosity can cause flooding, where the wick becomes saturated with liquid,

impeding capillary action and heat transfer. On the contrary, lower porosity can lead to limited capillary action but can reduce the risk of flooding as lower porosity wicks have fewer void spaces to fill with liquid. As a result, an ideal wick should have wick porosity large enough to sustain capillary pumping but not so large as to flood the entire wick. Sintered powder metal wicks achieve this balance nicely provided the grain size is selected appropriately.

- When constructed using appropriately chosen metals, they can have large thermal conductivities
- These kinds of wick are typically more efficient and reliable in heat transfer

Some disadvantages include being expensive to manufacture, as the transformation of metal to powder is involved in the process of making sintered powder metal wicks.

Grooved wick

Grooved wicks are formed of channels or grooves carved into the interior surfaces of the heat pipe. These grooves facilitate capillary action and surface tension forces, helping in the flow of the working fluid within the heat pipe. Grooved wick heat pipes are used in the electronic industry, the space industry and in energy systems. Some advantages of this kind of wick design include

- Grooved wicks can provide excellent heat transfer performance, in high heat input application
- This kind of wick is compatible with an especially wide range of working fluids.

Wire screen mesh wick

Wire screen mesh wicks are typically made from woven or knitted metallic wires, forming a porous mesh structure. The interconnected pores help transport fluid from the condenser to the evaporator section. Similar to grooved wicks, wire screen mesh wicks are also applicable in electronic cooling applications, space applications, and are likewise used for energy system cooling. Some advantages of wire screen mesh wicks are their high porosity, mechanical strength and durability. Disadvantages include a susceptibility to mesh corrosion or degradation, which underscores the importance of selecting a compatible working fluid. Also, the manufacturing of these wicks is not easy and can be expensive.

Fiber wick

Fiber wicks are composed of fine fibers arranged in a porous structure made of materials such as carbon, ceramic, or metal. The manufacturing process includes weaving, knitting, or directly depositing fibers onto a substrate. The major advantage of fiber wicks is their flexibility and ability to be modelled into different shapes and configurations. However, manufacturing complexity is a limitation of this kind of wick structure.

2.5.2 Working Fluid

A working fluid transfers heat by changing the phase from vapor to liquid and vice versa. The selection of working fluid depends on several factors. First and foremost is the compatibility with the wick structure and material of the heat pipe because the wettability of the wick structure impacts the performance of the heat pipe e.g. by changing the mode of condensation from filmwise to dropwise. Additionally, if the working fluid is incompatible with the material or wick material of the heat pipe, it can corrode or degrade the heat pipe, which eventually causes the heat pipe to fail as a vehicle for heat transfer. A potential working fluid must have high latent heat, high thermal conductivity, low liquid and vapor viscosity, and high surface tension. High surface tension enables the working fluid to generate high capillary forces. In the case of heat pipes with a wick structure, a working fluid with high wettability is desired because the working fluid will uniformly wet the wick. Meanwhile, working

fluids having high latent heat are, on a per unit volume basis, associated with more heat transfer. Low liquid and vapor viscosity is desired because having lower liquid viscosity can aid in the liquid return process to the evaporator.

2.5.3 Heat pipe material

The selection of the material for the heat pipe depends on its thermal conductivity, porosity, wettability, and ease of fabrication. Nonporous materials are favored to avoid vapor diffusion. A thermally conductive material is likewise favored to maximize heat transfer with the heat source and heat sink. Wettability is crucial for wickless heat pipes because the mode of condensation and, therefore, the rate of liquid return to the condenser depend on the wettability of the internal surface. This is why the application of a surface chemical coating to modify the surface wettability is a topic of keen interest including in the present study. If the wettability of the internal surface is high, condensate will appear in the form of a film, however, if the wettability is low then the condensate will appear in the form of discrete droplets. The condensation process is explained in detail in chapter 3 section 3.2.

2.6 Studies on surface modification of heat pipes

Here we summarize the available literature related to heat pipe surface modification. Research into this subject began in 1920 when Schmidt and his research team discovered the potential of dropwise condensation in the heat transfer process [44]. In the years since Schmidt's pioneering work, much further study has been devoted to the topic of surface modification and the techniques necessary to achieve such modifications. These techniques include electrochemical coating, hydrophilic and hydrophobic coating, simple immersion technique, chemical vapor deposition, physical vapor deposition, microporous coating, indium coating, polymer-based coating, anodization, electrospinning, etc. [22].

Xue et al. [62] recently conducted an experimental and numerical study that investigated the effect of surface chemistry in a flat plate heat pipe. Chemical etching of an aluminum surface was applied to create micro nanostructures. NaOH was used in the etching process followed by HCl. After the etching step was completed, the surface became hydrophilic. The surface was then sprayed with a polydimethylsiloxane mixture to render the surface hydrophobic. Compared to a base case devoid of chemical treatment, measured results indicated lower thermal resistances in cases where the evaporator was hydrophilic, and the adiabatic and condenser sections were hydrophobic. Simulation results indicate that the hydrophobic surface improved the droplet return rate and convective heat transfer coefficient. This study by Xue et al. [62] has given potential insights into surface coating applications for better heat transfer performance. However, experimental observations did not include qualitative descriptions of the condensation mode because the heat pipe apparatus did not permit visual access. In another experimental study by Zhang et al. [63], the coating of a high thermal conductive metal had been applied on a low thermal conductive carbon felt in a loop heat pipe. They investigated the coating of copper, zinc, and aluminum separately on the carbon felt of the loop heat pipe. The results indicated that aluminum and zinc coatings increase the thermal conductivity of the wick and the evaporator surface and also enhance the capillary force. However, in the case of copper coating, the trend does not fit. This is because the copper-coated wick, with its high thermal conductivity, causes increased heat transfer to the compensation chamber during start-up which is not the case for the aluminum and zinc coatings. This rise in temperature impedes the circulation of the working fluid, resulting in a slower start-up speed for the heat pipe. Figure 2.5 shows the thermal resistance of different high thermally conductive metal coatings compared with their non-coated analogues.

Another study performed by Zhan et al. [64] explored the possibility of modi-

fying the surface wettability so as to enhance the heat transfer performance of a gravity-driven heat pipe. The simulation was done using a gas-liquid model of the pseudo-potential lattice Boltzmann method. The simulation results were used to explain experimental results conducted by the same authors. For the experiments, chemical etching of the copper base surface was achieved using a $FeCl_3$ solution in the preparation of hydrophilic surface. To increase hydrophobicity, surface energy was lowered by treating the surface with octadecyl mercaptan. The study suggests that the hydrophobicity of the condensing surface enhances the heat transfer performance because as hydrophobicity increases, the liquid film of the condensed liquid becomes thinner and easier to detach because it forms condensate droplets instead of a condensate film and thus benefits the liquid return process. Although the numerical and experimental results suggest the benefit of having a hydrophobic condensing surface, no photos or videos illustrating the condensation process were presented. In another numerical simulation study performed by Wang et al. [65], they concluded that having a superhydrophobic condenser section and a superhydrophilic evaporator section enhances heat pipe performance by 3.03 to 7.99%. The reason was that a superhydrophobic condenser exhibits dropwise condensation, which enables a faster liquid return to the evaporator, while a superhydrophilic evaporator increases boiling in the evaporator section. The combination of both effects served to increase the overall rate of heat transfer.

The studies mentioned in the section 2.6 considered the effect of surface modification on the overall heat transfer process of a heat pipe. However, there are other parameters that can impact the heat transfer behavior along with the modified surface. In a relevant experimental study performed by Kim et al. [66], a two-phase closed thermosyphon was used to experimentally explore the combined effects of surface modification and fill fraction. It was concluded that rendering the condenser hydrophobic had a maximum effect at high fill fractions.



Figure 2.5: Overall thermal resistance of flat plate heat pipe with highly conductive metal coating (From [63]; used with permission.)

Although there are several numerical and experimental studies that favor surface modification to increase thermal conductivity, other studies offer a different point of view. In a numerical simulation study by Deng et al. [67], molecular dynamic simulations explored the possibility of modifying surface patterns within the condenser section of a nanoscale heat pipe. Results indicate that introducing the modified pattern on the cold surface does not significantly impact the heat transfer rate. Based on the simulation results, the authors suggest that improving condensation by fabricating nanopillars on the condensing surface of a nanoscale heat pipe may not necessarily increase the heat pipe thermal conductivity because the nanostructures can potentially restrict the working fluid circulation, which thereby negates any positive effects related to the larger condensation rate.

On the basis of this previous discussion, it seems like more investigation is needed to get conclusive outcome whether surface chemical treatment can potentially benefit the heat transfer performance of a heat pipe, which is why we are conducting experiments in this study which will relate to the condensation behavior of chemically coated surfaces to the overall heat transfer of a representative flat plate heat pipe.

2.6.1 Studies having visual access inside of the heat pipe

A number of experimental studies emphasize the importance of having visual access inside the heat pipe. In the study by Liao et al. [68], for instance, the researchers studied an ultra-thin flat-plate heat pipe with a sintered powder wick under various heating power inputs. Liao et al. observed three stages of phase change at different heat input conditions. When the heat input was less than 7 W, the observed phase change phenomena was evaporation which is shown in figure 2.6. The authors concluded that heat was transferred to the liquid-vapor interface through the copper plate where film evaporation occurs. The next stage was the buuble stage in medium heat input cases, where bubbles were observed to grow and datach flowing through the liquid-vapor interface into the condenser section. Figure 2.7 demonstrates the bubble stage observed at 10 W and 15 W of heat input. The third stage was the continuous vapor flow stage, which was at high heat input. In the figure 2.8, it was observed that there was almost no working fluid in the evaporator, which means the boiling limit was reached. This particular study shows the different stages of phase change as a function of different heat input conditions from lower to higher heat input values. Zhang et al. [69] performed a high-speed camera visualization experiment to study flow pattern variations in a pulsating heat pipe (PHP) containing a nanofluid. Their findings indicate that including nanoparticles encourages the phase transition of the working fluid within the PHP. These effects facilitate the return flow of condensed liquid and mitigate the occurrence of dry-out, which is defined as a state where there is no working fluid in the evaporator region to transfer heat by phase change to vapor.

In another experimental study by Malla et al. [70], the effect of surface chemistry in a compact flat plate heat pipe and flat plate pulsating heat pipe is explored. Results



Figure 2.6: Evaporation stage phase change phenomena observed at 5 W of heat input (From [68]; used with permission.)

show a maximum decrease of 42% in thermal resistance between modified and unmodified surfaces. This study concludes that applying a superhydrophobic coating in the condenser of the compact flat plate heat pipe drastically enhances the heat pipe performance. On the other hand, in the case of a flat plate pulsating heat pipe exhibiting superhydrophilic surfaces in the evaporator, adiabatic and condenser sections, there is a 60% decrease in the thermal resistance. The compact flat plate heat pipe did not permit any kind of visual access but the flat plate pulsating heat pipe apparatus did. Figure 2.9 shows the flow visualization of the flat plate pulsating heat pipe. The untreated heat pipe from the far left-hand side shows droplet oscillation, and the superhydrophilic treated heat pipe shows slug plug oscillations. Meanwhile, the case of a hydrophilic evaporator and superhydrophobic adiabatic and condenser section case shows evaporator dry-out. Lastly the superhydrophobic treated case shows Taylor droplet oscillation. This visualization study concludes that having superhydrophilic surface is beneficial in case of a flat plate pulsating heat pipe.



Figure 2.7: Bubble stage phase change phenomena observed at (a) 10 W and (b) 15 W of heat input (From [68]; used with permission.)

2.7 Conclusion and outlook

As we compare and compile all the concluding remarks of the above-mentioned studies we glean the following set of insights:

- Hydrophobic condenser sections can potentially increase heat pipe performance [62, 64, 65, 70].
- If the nanostructure is created in abundance to achieve superhydrophobic surface wettability in a nanoscale heat pipe, the free flow of liquid might be im-



Figure 2.8: Continuous vapor flow phase change phenomena observed at (a) 20 W and (b) 25 W of heat input (From [68]; used with permission.)

peded. As a result, it is important to explore different surface chemical modifications or surface chemical coatings to determine which coating in the condenser section best improves condensation and liquid return the most [67].

• A superhydrophilic evaporator section is beneficial because such a surface chemical regime exhibits greater bubble cores of boiling in the evaporator [70].

While the studies mentioned above offer valuable insights and intriguing observations regarding the use of modified surfaces to enhance heat pipe performance, to the



Figure 2.9: Flow visualization of a flat plate pulsating heat pipe with various wettability combinations: Bare represents untreated, SHPL represents superhydrophilic treated, HPL+SHPO represents a hydrophilic evaporator and superhydrophobic adiabatic and condenser sections, and SHPO represents superhydrophobic treated : the scale bar on the top of each image is 1 cm (From [70]; used with permission.)

best of our knowledge, there has been limited experimental study of the condensation behavior of chemically coated surfaces that relates to the consequent impact in the overall heat pipe performance in a heat pipe application. Indeed, what work has been completed often emphasizes qualitative observation at the expense of quantitative measurement or else quantitative measurement at the expense of qualitative observation. Our study aims to address this research gap by blending qualitative observation and quantitative measurement so that a holistic characterization of the impact of applying a surface chemical treatment can be obtained. To this end, we are especially interested in establishing a correlation between condensation performance, as indicated by droplet shedding behavior on various chemically treated surfaces, and the overall performance of a representative flat plate heat pipe.

Chapter 3 Condensation chamber experiment

3.1 Abstract

This chapter provides a detailed description of the condensation chamber experiments. It also specifies the approach followed in fabricating the samples. We include and discuss a comparative data plot to contrast the condensation behavior of the prepared samples. The footage from condensation chamber experiments was processed to extract quantitative information on the condensation behaviour of the samples. The experimental data collected from the condensation chamber indicates that applying a surface chemical coating promotes liquid shedding from the condensing surface compared to untreated samples.

3.2 Introduction

There are two modes of condensation depending on the nature of the condensing surface's wettability: filmwise condensation (FWC) and dropwise condensation (DWC). In filmwise condensation, the condensate is deposited on the condensing surface as a film. This film acts as an insulator in the process of heat transfer because it serves as a barrier (or additional resistance) between the heat source and sink. In dropwise condensation, by contrast, the condensate is deposited in the form of small droplets produced on micro-nano defects of the surface, which is called the nucleation site. The droplets progressively grow in size in time as more and more liquid accumulates along then within the droplet boundary. Except for horizontal substrates, droplets roll off the condensing surface when they grow large enough to overcome surface adhesion forces. Because of the rolling off of the drop, there is a new nucleation site formed where the process of droplet formation and growth can repeat. This repetition of nucleation and condensation due to the departure of condensate provides an advantage for the process of heat transfer relative to filmwise condensation for which the process of liquid shedding, though continuous, is generally much slower. As a result, heat transfer during the condensation process depends on the mode of condensation and the wettability of the condensing surface.

A surface is considered hydrophilic when a water droplet has a surface contact angle less than 90°. Contact angles below 10° are further classified as corresponding to a superhydrophilic surface. On the other hand, surfaces exhibiting contact angles greater than or equal to 90° are referred to as hydrophobic surfaces, while contact angles surpassing 150° correspond to superhydrophobic surfaces [71, 72]. An illustration of different wettability regimes is shown in figure 3.1. The wettability of a surface is an important consideration for mode of condensation because one of the ways of promoting dropwise condensation is by increasing the hydrophobicity of the condensing surface, which will promote droplet roll-off [73] and so increase the overall rate of heat transfer by up to ten times [74]. To promote dropwise condensation on a condensing surface, researchers have followed different surface fabrication processes based on the material composition or properties.

Dropwise condensation is promoted by making surfaces superhydrophobic by increasing surface roughness as a result of introducing micro-nano structures on the surface and then modifying the surface to lower the surface energy which increases the hydrophobicity [75, 76]. By the combination of micro-nano structures and the lowering of surface energy, the degree of superhydrophobicity increases [77]. The approach taken in this study utilizes a surface chemical etching process for the creation of micro-nano structures and the lowering of surface energy [75]. A flow diagram of all the significant experimental steps taken in this study for the condensation chamber experiments is illustrated in figure 3.2.



Figure 3.1: Wettability regimes of surface with different surface contact angle represented by θ



Figure 3.2: Experimental flow diagram of condensation chamber experiments

3.3 Surface chemical treatment

3.3.1 Experimental steps

Materials used

Mirror-like multipurpose 110 copper sheeting was used for copper sample preparation and polished multipurpose 6061 aluminum was used for aluminum sample preparation. Potassium hydroxide (KOH), Sodium Hydroxide (NaOH), ethanol (C_2H_5OH) and lauric acid $(C_{12}H_{24}O_2)$ all purchased through Fisher Scientific were used in the chemical treatment process. Details of the materials and chemicals used in the preparation of the samples are given in table 3.1.

Table 3.1: N	Material and	chemicals t	that w	vere used	in t	he prepara	ution of	samples
--------------	--------------	-------------	--------	-----------	------	------------	----------	---------

Material and chemical list for surface coating preparation				
Material/Chemicals	Supplier	Details		
Copper sheet	McMaster-Carr	Mirror-like multipur- pose 110 copper sheet		
Aluminum sheet	McMaster-Carr	Polished multipurpose 6061 aluminum sheet		
Potassium hydroxide (KOH)	Fisher Scientific	Potassium hydrox- ide (pellets/certified ACS), 500 gm		
Ammonium persul- phate	MG Chemicals	410 Ammonium persulphate copper etchant, 1 Kg solid crystals		
Sodium hydroxide (NaOH)	Fisher Scientific	Sodium hydroxide (pellets/certified ACS), 1 kg		
Ethanol (C_2H_5OH)	LabChem	Reagent ethanol, ACS grade, 94.0% to 96.0 $\% (v/v)$		
Lauric acid	TCL America	Lauric acid 98.0+%		

Sample and solution preparation: Aluminum

Samples of dimension 25.4 mm \times 25.4 mm \times 3 mm (Length \times Breadth \times Thickness) were machined using a waterjet cutter from the stock material, which was received from McMaster-Carr. The samples were cleaned using distilled water. 100 ml of etching solution was prepared by dissolving potassium hydroxide (KOH) pellets in deionized (DI) water. The requisite mass of potassium hydroxide was measured using

a precision balance (Sartorius 1265 MP). A separate 100 ml of solution for the lowering of surface energy was prepared by dissolving lauric acid into ethanol – details, including solution concentrations, are provided in the appendix.

Sample and solution preparation: Copper

For copper, all of the above steps were the same except that the etching solution was prepared by dissolving 10 g of sodium hydroxide pellets in 100 ml of DI water along with 2.281 g of ammonium persulphate as a catalyst. Additional chemical details are provided in the appendix.

Chemical treatment process

Samples were immersed into the etching solution for different durations, ranging from 30 to 150 minutes for the etching process [75, 76]. After the etching process, the samples were removed from the etching solution and gently cleaned using ethanol and DI water. The etched samples were dried for 60 minutes and kept inside a fume hood. In the second chemical treatment step, dry samples were immersed in the lauric acid/ethanol solution to lower the surface energy by creating a sponge-like layer on the etched surface [75, 76]. The immersion time was 60 minutes. Thereafter, samples were again cleaned with ethanol and dried in the fume hood for 60 minutes. In this fashion, different samples were prepared based on etching time and on the concentration of the etching solution. Details concerning these latter two parameters are given in table 3.2 and 3.3. Meanwhile, a schematic of the chemical process is given in figure 3.3.

Chemical reaction: Aluminum sample

In the first step of the chemical treatment process, aluminum or a layer of aluminum oxide present on the aluminum sample reacts with an aqueous solution of potassium hydroxide (KOH) and forms potassium aluminate ($K[Al(OH)_4]$) which is soluble in



Figure 3.3: Surface chemical treatment process for both copper and aluminum samples (the second process step is the same for both aluminum and copper).

KOH solution. The chemical reactions are shown below

$$Al_2O_{3(s)} + 2KOH_{(aq)} + 3H_2O = 2K[Al(OH)_4]_{(aq)}$$
 (3.1)

$$2Al_{(s)} + 2KOH_{(aq)} + 6H_2O = 2K[Al(OH)_4]_{(aq)} + 3H_2$$
(3.2)

In the second step of the chemical reaction, the carboxyl group of lauric acid reacts with the aluminum atom of the sample through the process shown below:

$$Al^{3+} + 3CH_3(CH_2)_{10}COO^- = Al(CH_3(CH_2)_{10}COO)_3$$
(3.3)

Due to bonding by the long negative end of the alkyl to the aluminum atom, an aluminum sample with low surface energy is formed [75].

Sample name	Etching time (mins)	Concentration of etching solution (molarity)	Contact angle θ (°)	Standard deviation (°)	Remarks
Al S 0	0	Nil	85.29	3.37	Untreated
Al S 1	30	0.18	117.97	3.21	Hydrophobic
Al S 2	45	0.18	136.59	0.81	Hydrophobic
Al S 3	40	1.5	154.35	1.31	Super- hydrophobic
Al S 4	45	1.5	166.10	1.87	Super- hydrophobic

Table 3.2: Results of the surface chemical treatment applied to aluminum (Al) samples

Chemical reaction: Copper sample

In the first step of the chemical treatment process, $(NH_4)_2S_2O_8$ and NaOH reacts with and oxidizes copper. By this process of dissolution, etching of the copper sample occurs. The associated chemical reactions are given below:

$$Cu^{2+} + 2OH^{-} = Cu(OH)_2$$
 (3.4)

$$4Cu + 14OH^{-} + 4(NH_4)_2S_2O_8 = Cu_4(SO_4)(OH)_6 + 7SO_4^{2-} + 8NH_3 + 8H_2O \quad (3.5)$$

In the second step of the chemical reaction, the carboxyl group of lauric acid reacts with the copper atom through the process shown below:

$$Cu^{2+} + 2CH_3(CH_2)_{10}COO^- = Cu(CH_3(CH_2)_{10}COO)_2$$
(3.6)

Due to bonding by the long negative end of the alkyl to the copper atom, a copper sample with low surface energy is formed [76].

Sample name	Etching time (mins)	Concentration of etching solution (molarity)	$\begin{array}{c} \text{Contact} \\ \text{angle} \theta \\ (^{\circ}) \end{array}$	Standard deviation (°)	Remarks
CuS0	0	Nil	58.03	2.88	Untreated
CuS1	30	2.5	144.17	2.98	Hydrophobic
CuS2	60	2.5	147.49	0.42	Hydrophobic
CuS3	90	2.5	152.79	0.48	Super- hydrophobic
CuS4	120	2.5	158.07	1.81	Super- hydrophobic
CuS5	150	2.5	167	0.5	Super- hydrophobic

Table 3.3: Results of the surface chemical treatment applied to copper (Cu) sample

3.4 Condensation chamber experiment

3.4.1 Experimental setup design

A schematic of the experimental setup along with the CAD drawing of the condensation chamber are shown in figure 3.4. The experimental setup consists of a Peltier unit, condensation chamber housing, humidifier, and data acquisition system. The Peltier unit primarily consists of a Thermoelect Assy Direct-Air 44 W and an aluminum thermocouple adapter, which is attached to the Peltier surface. The housing includes a front enclosure that has a humid air inlet and a viewing window. A humidifier was attached to the humid air inlet of the front enclosure using a pipe. During the experiment, humid air flowed inside the condensation chamber through this pipe. Samples were placed on the thermocouple adapter of the Peltier unit. In between the Peltier unit and the sample, four thermocouples were placed, which measured the sample temperature at different points in space. As the thickness of the samples was 3 mm and the samples are characterized by relatively large thermal conductivities, it was safely assumed that the surface temperature should be close to the temperature measured by the thermocouples. The thermocouples are T type thermocouples (5SRTC-TT-T-20-36, OMEGA) and have a resolution of 0.1°C and an accuracy of $\pm 1^{\circ}$ C. To prevent electrical short circuits caused by direct contact between the sample surface and thermocouples, thermally conductive and electrically insulating tape (8810, 3M) was attached on the tip of each thermocouple. Thermal paste (corsair XTM50) was applied in between the thermocouple adapter and the back of the sample to reduce thermal resistances associated with a possible air gap and, by extension, to decrease temperature non-uniformities on the exposed surface of the sample. There was also a separate thermocouple placed in front of the humid air inlet that measured the ambient temperature. The difference between the ambient temperature and the sample temperature was defined as the subcooling temperature. The thermocouples were calibrated in an ice water bath using a Fluke 54-2 B dual-input data logging thermometer (52-2, FLUKE). The deviation of the thermocouple was $\pm 0.82^{\circ}$ C. The temperature data captured by the thermocouples was collected by a custom-made data acquisition (DAQ) system. Temperature measurements were accessed through the serial monitor integrated into an Arduino IDE on a personal computer. The Peltier was controlled by Laird Thermal Software on a personal computer.

Visual recording using a digital single-lens reflex camera (Nikon D7100) with Nikkor AF-S 105 mm 1:2.8 G ED lens and Nikon AF-S 20EIII 2X Teleconverter lens was performed. Images were collected through the viewing window on the front enclosure.

3.4.2 Experimental steps

The experimental steps were as follows:

- 1. The sample was placed on the Peltier unit.
- 2. The humidifier was activated causing a flow of humid air to enter the condensa-



Figure 3.4: Condensation chamber setup top view and exploded CAD model.

tion chamber. Simultaneously the camera recording was started and the Peltier unit and the DAQ system were activated.

- 3. The experiment was run for a total of 20 minutes.
- 4. All the while, a petri dish of 100 mm diameter was placed underneath the sample inside the condensation chamber during the experiment to collect the droplets that rolled down and were subsequently discharged from the condensing surface. By measuring the mass of water collected in this dish, the average rate of water discharge could be determined.

3.4.3 Image processing

Image processing was performed to quantify the number of droplets condensed on the cold surface and to compare patterns of droplet shedding between the untreated and super hydrophobically treated samples. Image Pro Premier software (version 9.3) from Media Cybernetics was used for image analysis. Note that we did not quantitatively compare rates of droplet shedding with the rate of condensation as determined by measuring the mass of water collected in the petri dish. Doing so would have required detailed information regarding not only the droplet diameter but also its three-dimensional shape.

Image processing steps were as follows:

- The experimental video was imported as a series of image sequences in Image Pro Premier software.
- 2. For calibration purposes, a needle was placed in the field of view while performing the experiments. The needle diameter was measured, and the resulting measurement was used to convert from pixel coordinates to physical coordinates in the image processing software.

- 3. After calibration, a region of interest (ROI) was carefully selected based on the homogeneity of droplet nucleation and light intensity of the condensing surface.
- 4. Experimental images were extracted at 1 minute intervals giving 21 images total for an experiment lasting 20 minutes.
- 5. All the images were converted into monochrome.
- 6. Manual thresholding was used for segmenting droplets, and the count function was used to determine the number of droplets in any given experimental frame. The data extracted from each frame, therefore, included droplet count as well as mean radius (mm).

An example of the approach taken in image processing is explained in detail in the appendix .

3.5 Results and discussion

3.5.1 Sample characterization

The water contact angle of each of the samples was measured using the Mobile Surface Analyzer (MSA, Krüss). The footages taken by the Mobile Surface Analyzer were analyzed using Advanced software from Krüss. SEM images were also taken using a (Zeiss EVO MA10) scanning electron microscope from the nanoFAB facility of the University of Alberta.

Figures 3.5 and 3.6 show the sessile drop test footage for representative samples spanning a range of contact angles. Figures 3.7 and 3.8 show SEM images of representative samples spanning a range of contact angles. From the SEM images of aluminum in figure 3.7, it is observed that as the etch time is increased and the concentration of etching solution is increased, the number of micro-nano structures likewise increases, which in turn increases the hydrophobicity. Similar conclusions apply in case of copper – see figure 3.8. By varying etch time and solution concentration, a total of two

hydrophobic and two superhydrophobic aluminum samples are fabricated, and two hydrophobic and three superhydrophobic copper samples are fabricated.



Figure 3.5: Aluminum sample sessile drop test images taken using the Mobile Surface Analyzer (MSA, Krüss): the θ values of each sample are given in parenthesis.

0.5 mm (a) Cu S 0 (58.03°)	(b) Cu S 1 (144.17°)	(c) Cu S 2 (147.49°)
(d) Cu S 3 (152.79°)	(e) Cu S 4 (158.07°)	(f) Cu S 5 (167.00°)
0		0

Figure 3.6: Copper sample sessile drop test images taken using the Mobile Surface Analyzer (MSA, Krüss): the θ values of each sample are given in parenthesis.

3.5.2 Water discharge

At the conclusion of each experiment, the mass of water discharged by the condensing surface and collected in the petri dish was measured; figures 3.9 and 3.10 show the corresponding data where the mass of water has been normalized by the surface area of the condensing surface and time. The different curves represent different subcooling temperatures, i.e. 5°C, 10°C, 15°C and 19°C.

From figures 3.9 and 3.10, it is observed that as we increase the hydrophobicity of the surface, droplet roll-off is enhanced. This trend applies to both copper and



Figure 3.7: Aluminum sample SEM images with the corresponding sessile drop test images: the θ values of each sample are given in parenthesis (the scale bars in the first subset image are the same for all other subset images)



Figure 3.8: Copper sample SEM images with the corresponding sessile drop test images: the θ values of each sample are given in parenthesis (the scale bars in the first subset image are the same for all other subset images)

aluminum. Also, and as expected, the water discharge increases as we increase the subcooling temperature from 5°C to 19°C.

3.5.3 Image processing (aluminum)

Figures 3.11 (a),(b) respectively show time series data for samples Al S 0 and Al S 4. Included in the figures are average droplet radius data and droplet count data. The figures in question indicate that the number of droplets is larger for Al S 4 (contact angle of 166.10°) than it is for Al S 0 (contact angle of 85.29°) because of the higher surface hydrophobicity. Correspondingly, droplets that appear on sample AL S 4 are, on average, smaller than those that appear on sample AL S 0. This difference



Figure 3.9: Water discharge vs. surface contact angle for the aluminum samples. Subcooling temperatures are as indicated.



Figure 3.10: Water discharge vs. surface contact angle for the copper samples. Subcooling temperatures are as indicated.

of size is because of the lower surface energy. A comparison of frames 2 mins, 6 mins and 14 mins are shown in figure 3.11 (c). The 2-mins frame represents the

initial nucleation of droplets in both samples. Until 2 minutes into the experiment, the number of droplets nucleated in Al S 0 is greater than in Al S 4. However, the nucleated droplets stay on the condensing surface because the surface has higher surface energy. The droplets start to coalesce with neighboring droplets and then grow in size, which explains why the number of droplets starts to decrease after 2 minutes for Al S 0. There is a sudden drop in the average droplet radius for Al S 0 after 13 mins of experimental run-time. This is because a large droplet rolled off the condensing surface leaving in its wake fresh solid surface that was quickly nucleated with a myriad of small droplets. By contrast, for the case of Al S 4, the droplet count was more or less stable after 5 minutes of experimental run time. This is because there is a near-constant balance of droplet roll-off and renucleation over the duration of the experiment. Correspondingly, we find that the last two of the experimental images from figure 3.11 (c) look very similar in appearance.

3.5.4 Image processing (copper)

In like fashion to the above discussion, the results of droplet count and average droplet radius as extracted from experimental images of Cu S 5 (contact angle of 167.00°) are plotted in the figure 3.12 (a). The x-axis represents time, the left y-axis represents the average droplet radius and the right y-axis represents the droplet count. In contrast to surface chemically-treated, untreated samples such as Cu S 0 (contact angle of 58.03°) exhibit a hydrophilic character and the condensation mode is therefore filmwise. As a result, the number of droplets and average droplet radius are not plotted in the figure 3.12. A comparison between frames collected after 2 mins and 14 mins for the copper samples Cu S 0 and Cu S 5 is shown in figure 3.12 (b). This comparison illustrates the clear difference between the two modes of condensation relevant to Cu S 0 and Cu S 5. In the former case, the condensate film serves as an additional thermal resistance, i.e. the heat transfer process is impeded. This explains why we see less liquid water



Figure 3.11: (a) Condensation chamber footage analysis for Al S 0 (contact angle of 85.29°) (left *y*-axis: average droplet radius, right *y*-axis: droplet count, both plotted against time) (b) Condensation chamber footage analysis for Al S 4 (contact angle of 166.10°) (left *y*-axis: average droplet radius, right *y*-axis: droplet count, both plotted against time) (c) Experimental images taken at 2 min, 6 min and 14 min of experimental run time. Top panel: Al S 0; bottom panel: Al S 4.

discharge in figure 3.10 for Cu S 0 (contact angle of 58.03°) than we do for any of the other copper samples. More generally, the quantitative information extracted using image processing supports the conclusion evident from figures 3.9 and 3.10, namely that larger contact angles are associated with more robust dropwise condensation.



(b)

Figure 3.12: (a) Condensation chamber footage analysis for Cu S 5 (contact angle of 167.00°) (left *y*-axis: average droplet radius, right *y*-axis: droplet count, both plotted against time) (b) Experimental images taken at 2 min and 14 min of experimental run time. Top panel: Cu S 0; bottom panel: Cu S 5.

3.6 Conclusion

The purpose of performing the condensation chamber experiments was to examine the effect of applying a surface chemical treatment on the condensation behavior. As a result, small substrates were prepared to have different surface contact angles in order to examine their propensity for droplet shedding in the small-scale condensation chamber exhibited schematically in figure 3.4. The outcome of the condensation chamber experiment are summarized as follows:

• The mass of water droplets discharged from the condensing surface increases

with the surface contact angle for both aluminum and copper. Because having a superhydrophobic surface benefits droplet roll-off and subsequent renucleation, we expect and observe more droplets to be shed at larger contact angles.

Although our condensation chamber experiments confirm that applying a surface chemical treatment leads to more droplet shedding and more condensation overall, it remains to determine whether having a superhydrophobic surface coating in the condenser section of a heat pipe will or will not enhance performance This open question is explored in Chapter 4, which presents new experiments meant to "upscale" the laboratory investigation presented in this chapter.

Chapter 4 Optical heat pipe experiment

4.1 Abstract

This chapter presents a detailed description of the representative flat plate heat pipe experiments. Within the results section, a comparative plot is included that assesses the overall thermal resistances of heat pipes with and without chemical coatings applied to the condenser section From this analysis, we find that applying surface chemical coatings in the condenser section generally enhances overall heat transfer performance in certain instances. However, the effect is minimal or even negative in regions of the parameter space that we characterize in more detail below.

4.2 Introduction

In the last chapter we found that application of chemical coating can influence the rate of water shedding. To reiterate, the goal of this chapter is to explore whether increasing this rate of water shedding will or will not increase the overall heat transfer in a heat pipe. The answer to this question hinges on whether the rate of liquid return from the condenser to the evaporator is/is not a major limiting factor in the operation of the heat pipe. To gain the associated insights, we have designed a representative flat plate heat pipe, which is termed an "optical heat pipe." Before detailing the experimental results derived from this optical heat pipe, it is first necessary to describe
its design and fabrication. This is the topic of the following section.

4.3 Experimental setup

4.3.1 Heat pipe specification

The design of the optical heat pipe consisted of multiple layers starting from the bottom with a metal layer consisting of aluminum or copper. Thereafter, we included two polycarbonate layers (one transparent, the other not), and a reinforcing layer of aluminum, which formed the top-most layer As shown in figure 4.1, the assembly included two o-rings, which were fitted into grooves carved into both sides of the middle polycarbonate layer. The layers and o-rings were all bolted together so as to form a liquid water-tight enclosure. The different sections of the heat pipe with proper dimensions are shown in figure 4.1.

4.3.2 Setup Design

Our experimental setup included the optical heat pipe, a table with a tilting mechanism, a power supply, a chiller, a data acquisition system, and a computer capable of recording data. The evaporator section consisted of a rectangular-shaped block of length 10.16 cm and breadth of 5.08 cm into which four heating cartridges (26008, Tutko), each having an electrical resistance of 2 Ω , were inserted. All four heating cartridges were connected in parallel to a DC power supply unit (1687B, B&K Precision). The condenser section consisted of a rectangular block comprising of cooling channels, further attached to a chiller unit (9106A11B, Polyscience). The evaporator and condenser blocks were in thermal contact with the bottom metal surface of the optical heat pipe. An ethylene glycol/water solution was used as the chiller fluid where ethylene glycol and water were mixed in a volume ratio of 3:2. The heat pipe, along with the evaporator and condenser blocks, was placed on top of the tilting table, which allowed the heat pipe to be inclined through different angles ranging from $\alpha = -90^{\circ}$ to $\alpha = 90^{\circ}$.

Although we regard the heat pipe as being of "optical" variety, it is, for most of the duration of an experiment, covered with glass fiber insulation. The purpose of the insulation was to (i) decrease the time required to achieve steady state conditions, and, (ii) avoid condensation on the inside surface of the polycarbonate, which would have obscured experimental images. By removing portions of the insulation for brief periods of time, it was possible to collect experimental images, from which qualitative information concerning the nature of the condensation could be extracted. The collection and analysis of the images in question justifies our use of the term "optical heat pipe."

A schematic of the heat pipe experimental setup (without the insulation) is shown in figure 4.2.

The custom-made data acquisition system consisted of 10 T-type thermocouples (5SRTC-TT-T-20-36, OMEGA). These thermocouples had a resolution of 0.1°C and an accuracy of $\pm 1^{\circ}$ C. To prevent electrical short circuits caused by direct contact between the sample surface and thermocouples, thermally conductive and electrically insulating tape was attached on the tip of each thermocouple (8810, 3M). The thermocouples were calibrated in an ice water bath using a Fluke 54-2 B dual-input data logging thermometer (52-2, FLUKE). The deviation of the thermocouples were within $\pm 0.82^{\circ}$ C. The temperature data were collected through the serial monitor integrated into an Arduino IDE on a personal computer.

4.3.3 Design of experiments

The temperature gradient across the axial length of the optical heat pipe was recorded using the aforementioned series of thermocouples. The position of each of the thermocouples is shown in figure 4.1 (b). A total of four thermocouples were placed in the evaporator section, four thermocouples were placed in the adiabatic section and two



Figure 4.1: (a) Exploded schedmatic showing the arrangement of the different layers of the optical heat pipe. (b) Thermocouple position along the axial direction of the heat pipe T1 (0.30 cm), T2 (3.40), T3 (6.50 cm), T4 (9.60 cm), T5 (14.20 cm), T6 (22.20 cm), T7 (28.20 cm), T8 (36.20 cm), T9 (40.80 cm) and T10 (49.70 cm)

(b)

9.6

Axial Length (cm)

0.3

• 0

49.7

thermocouples were placed in the condenser section. The average of all four thermocouples in the evaporator section was considered as the evaporator temperature (T_e) . Similarly, for the condenser section the average of the two thermocouples was used as the condenser temperature (T_c) . The value of T_c was most strongly influenced by the chiller set point temperature, which we kept fixed at 0°C. Meanwhile, the value



Figure 4.2: Schematic diagram of the optical heat pipe experimental setup. Red arrows represent heat input and blue arrows represent heat removal.

of T_e was most strongly influenced by the heat input. In this study, the heat input, as measured from the voltage and current readings of the power supply unit, was set to 10 W, 20 W or 30 W. In a similar spirit, we considered three possible values for the heat pipe inclination angle, namely 0°, 3° and 90°. Experiments with $\alpha = 3^{\circ}$ represented a special case within our study. This particular inclination angle was based on the working fluid interface shape. When $\alpha = 3^{\circ}$ and the fill fraction is 100%, our calculations predict that the liquid-vapor interface will just intersect the boundary between the adiabatic and condenser sections. Thus, any droplets shed from the condenser will immediately form part of the liquid pool spanning the adiabatic section and, to a greater extent, the evaporator. Of course, when the fill fraction is less than 100%, condensed droplets will have to travel some distance across the adiabatic section before reaching this liquid pool. Therefore, and in greater contrast to the $\alpha = 0^{\circ}$



Figure 4.3: CAD model showing the optical heat pipe assembly. This schematic was generated using SOLIDWORKS 2023.

and 90° cases, $\alpha = 3^{\circ}$ provides a robust opportunity to gauge the relative importance of the droplet advection distance. Another variable parameter was the fill fraction, i.e. the volume of working fluid to the volume of the evaporator section.

Experiments were conducted both for untreated base plates as well as those that were chemically-treated to yield a superhydrophobic surface only in the condenser region of the heat pipe which is termed as "SH-treated" in this chapter. To compare the performance of SH-treated vs. untreated samples, a new performance indicator is introduced. This new variable is a nondimensional parameter that is given by (4.5).

$$T_e = \frac{T_1 + T_2 + T_3 + T_4}{4} \tag{4.1}$$

$$T_c = \frac{T_9 + T_{10}}{2} \tag{4.2}$$

$$\Delta T = T_e - T_c \tag{4.3}$$

$$R = \frac{T_e - T_c}{VI} \tag{4.4}$$

Performance Indicator = $\frac{(T_e - T_c)_{\text{untreated}} - (T_e - T_c)_{\text{SH-treated}}}{(T_e - T_c)_{\text{untreated}}} \times 100\%$ (4.5)

In the above equations, subscripts denote specific thermocouples as shown in figure 4.1 (b). R represents overall thermal resistance, V represents voltage, and I represents current.

4.3.4 Experimental steps

The experimental steps were as follows:

- 1. A specific volume of water was added through the evaporator-end valve using a syringe. During water addition, the condenser-end valve was kept closed. After the requisite volume of water was added, the evaporator-end valve was closed. The condenser-end valve was then connected to a vacuum pump (LABCONCO, Rotary Vane Vacuum Pump, 117 LPM, 115V). The purpose of using a vacuum pump was to evacuate any air present inside the optical heat pipe so that the heat pipe is left with only the desired amount of working fluid. The condenser-end value was opened, and the vacuum pump was turned on to achieve a relative vacuum pressure of 5kPa from the inside of the heat pipe. The vacuum pump was kept running for 20 minutes before the experiment was started. After 20 minutes, the condenser-end valve was closed, and the vacuum pump was turned off.
- 2. The DAQ card was connected to a computer, allowing us to collect readings from each of the 10 thermocouples every second.
- 3. The power supply and chiller were turned on.
- 4. After starting a particular experiment, the heat pipe was allowed to reach steady state, which is defined as a state where the measured temperatures of the evaporator and condenser change by no more than 0.1°C within a 10 minute interval.

- 5. After recording the temperature data, the experiment was stopped, and the setup was allowed to return to room temperature.
- 6. The above-mentioned steps were repeated three times for each combination of inclination angle, fill fraction and heat input.

4.4 Results and discussion

4.4.1 Inclination angle: $\alpha = 90^{\circ}$

The variation of overall thermal resistance, R with respect to fill fraction is plotted in figures 4.4 (a) and 4.4 (b). Figures 4.5 (a) and 4.5 (b) show the same data but for the case of the copper base plate. For both figures 4.4 and 4.5, the left- and right-hand side panels respectively consider untreated and chemically-treated specimens. From these figures, the following observations are apparent:

- 1. At 0% fill fraction, there is no working fluid inside the heat pipe. As a result the heat transfer mode is conduction Because the base plate thermal conductivity is diminished when a chemical coating is applied, the overall thermal resistance of the chemically-treated base plate is higher than the counterpart untreated base plate.
- 2. As the amount of working fluid is increased to and beyond 20% fill fraction, there is a decline in the R value (which is defined as the overall thermal resistance in equation 4.4): when more working fluid is introduced, heat pipe dryout is less prevalent.
- 3. After a certain fill fraction, the R value starts to increase due to evaporator section flooding. Because flooding is associated with an increase in the thermal resistance, the overall rate of heat transfer decreases leading to the increase in R.



Figure 4.4: R vs. fill fraction for (a) untreated and (b) SH-treated aluminum base plates In both cases, the heat pipe angle of inclination measures $\alpha = 90^{\circ}$.



Figure 4.5: R vs. fill fraction for (a) untreated and (b) SH-treated copper base plates In both cases, the heat pipe angle of inclination measures $\alpha = 90^{\circ}$.

4.4.2 Inclination angle: $\alpha = 0^{\circ}$

The variation of overall thermal resistance, R (°C/W) with respect to fill fraction (%) is plotted in figure 4.6 (a) and 4.6 (b) for aluminum heat pipe and figure 4.7 (a) and 4.7 (b) for copper when the heat pipe is in horizontal orientation to the ground. Similar trend to $\alpha=90^{\circ}$ is observed in the case of $\alpha=0^{\circ}$.



Figure 4.6: R vs. fill fraction for (a) untreated and (b) SH-treated aluminum base plates In both cases, the heat pipe angle of inclination measures $\alpha=0^{\circ}$



Figure 4.7: R vs. fill fraction for (a) untreated and (b) SH-treated copper base plates In both cases, the heat pipe angle of inclination measures $\alpha=0^{\circ}$

4.4.3 Inclination angle: $\alpha = 3^{\circ}$

If we analyze the trend of the thermal resistance curves, we see similar results to those encountered for the other two inclination angles. In figure 4.8 (a) and 4.8 (b) for aluminum and figure 4.9 (a) and 4.9 (b) for copper we can observe a reduction in overall thermal resistance to a certain fill fraction and after this certain fill fraction if the working fluid is increased either the overall thermal resistance remains same or the thermal resistance increases.



Figure 4.8: R vs. fill fraction for (a) untreated and (b) SH-treated aluminum base plates In both cases, the heat pipe angle of inclination measures $\alpha=3^{\circ}$



Figure 4.9: R vs. fill fraction for (a) untreated and (b) SH-treated copper base plates In both cases, the heat pipe angle of inclination measures $\alpha=3^{\circ}$

4.4.4 Comparison of performance of untreated and SH-treated heat pipe

Aluminum

Figure 4.10 shows a map of ΔT , defined as the difference between the evaporator and condenser temperatures at steady state – see (4.3). We consider in figure 4.10 each combination of inclination angle, fill fraction and heat input. Darker shades correspond to larger ΔT values and larger thermal resistances. Similar trends are observed for the SH-treated vs. untreated samples though, in the former case, the color shades within the map tend to be somewhat lighter. This difference is arguably most apparent when $\alpha=3^{\circ}$. Application of a surface chemical coating shows the most promise when the droplet advection distance is a minimum. The reason for the difference of ΔT values between untreated and SH-treated base plates can be explained using experimental images collected from the optical heat pipe. From figure 4.11, we observe that both untreated and SH-treated samples exhibit dropwise condensation owing to the fact that even untreated aluminum has a comparatively large contact angle. Thus the differences exhibited in the inset images to figure 4.11 are of degree rather than of kind: the lower surface energy of the SH-treated specimen yields many comparatively small droplets as compared to the smaller number of large droplets evident for the untreated specimen. Moreover, larger droplets are often pinned and therefore less mobile than the small droplets that collect along, but are more quickly shed from, SH-treated surfaces. This process of droplet shedding leads to the exposure of bare plate surface and subsequent renucleation. The net result of the aforementioned differences is to realize a greater rate of heat transfer when the base plate includes a chemical coating.

Copper

Data corresponding to figure 4.10 for the case of the copper base plate are shown in figure 4.12. Here, we observe generally lighter shades, especially when considering the chemically-treated specimens. These observations can again be explained with reference to the experimental images collected from the optical heat pipe. From figure 4.13 we note that the condensation mode for untreated copper is mostly filmwise, albeit with a smattering of droplets included. By contrast, SH-treated copper shifts to dropwise condensation. This difference in the mode of condensation is the primary reason that, in contrast to the aluminum case, chemically-treated copper base plates tend to outperform their untreated counterparts.

Non Treated		0° inclina	tion	39	° inclinati	on	90° inclination					
	10 W	20 W	30 W	10 W	20 W	30 W	10 W	20 W	30 W			
0 %	40.43	61.45	67.31	40.43	61.45	67.31	40.43	61.45	67.31			
20 %	28.80	47.70	40.05	32.15	49.80	46.95	32.65	52.00	51.45			
40 %	31.15	43.00	38.85	31.10	32.50	33.45	31.20	32.90	36.60			
60 %	32.55	32.20	42.00	30.50	45.10	40.65	31.35	40.50	42.15			
80 %	32.60	44.00	46.50	29.95	47.30	42.30	31.70	41.30	39.45			
100 %	33.50	53.70	49.50	30.15	49.30	43.80	32.00	40.70	39.15			
	(a)											

ΔT (°C)

80.00 75.00 70.00 65.00 55.00 55.00 45.00 40.00 35.00 30.00 25.00 20.00 15.00

SH Treated	0° inclination			3° inclination			90° inclination			
	10 W	20 W	30 W	10 W	20 W	30 W	10 W	20 W	30 W	
0 %	42.60	65.91	72.52	42.60	65.91	72.52	42.60	65.91	72.52	
20 %	28.95	34.20	39.75	30.60	45.10	45.30	31.20	46.10	40.95	
40 %	31.35	37.50	40.05	30.15	29.70	31.35	23.40	32.10	37.95	
60 %	31.50	34.30	43.50	29.80	43.10	39.30	29.00	35.70	38.40	
80 %	30.85	42.80	49.95	29.70	46.70	40.80	31.70	40.30	36.30	
100 %	31.85	51.60	48.30	29.50	46.90	42.60	34.05	40.40	37.80	
					(b)					

Figure 4.10: ΔT color map for aluminum. (a) Untreated and (b) SH-treated.

Performance indicator color plot

The discussion concerning figures 4.11 and 4.13 has largely focused on performance differences between SH-treated vs. untreated base plates. We can expand on this comparison by revisiting the parameter performance indicator, which is defined by (4.5). Using this definition and the data of figures 4.11 and 4.13, one can generate color maps of the type illustrated by figure 4.14. In figure 4.14, reddish and bluish colors respectively represent scenarios where application of a chemical coating is not



Figure 4.11: Experimental images taken from the optical heat pipe at steady state conditions for a heat input of 30 W, an inclination angle of $\alpha=90^{\circ}$ and various fill fractions. Aluminum base plate

vs. is beneficial. White regions are those where roughly equal performance is noted, i.e. the chemical coating has no pronounced positive or negative effect. Although the panels of figure 4.14 do not show an easily-identifiable overall trend, there are a number of important observations that follow from this figure, i.e.

 Initially, at 0 % fill fraction where there is no working fluid, the heat transfer mode is conduction. As a result, the material's thermal conductivity plays a major role in heat transfer. Because there exists an extra layer of chemical coating in the SH treated sample, the thermal conductivity is less than the untreated sample and the cells of figure 4.14 therefore appear red. By contrast,

(a)	0° inclination			3° inclination			90° inclination			
Non Treated	10 W	20 W	30 W	10 W	20 W	30 W	10 W	20 W	30 W	
0 %	36.15	54.61	63.55	36.15	54.61	63.55	36.15	54.61	63.55	
20 %	21.35	34.40	33.75	23.00	34.30	41.70	20.50	29.10	33.15	
40 %	19.60	29.20	24.90	22.3	28.90	37.50	20.20	23.60	32.40	
60 %	17.15	21.80	22.35	20.8	21.80	26.25	17.05	20.10	23.70	
80 %	21.75	21.80	30.90	15.00	21.90	24.60	20.70	21.40	25.80	
100 %	21.7	18.40	26.40	15.10	18.50	22.20	21.2	24.20	27.30	

(4)	(a)
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ΔT (°C)

80.00 75.00 70.00 65.00 55.00 55.00 45.00 45.00 35.00 35.00 25.00 20.00 15.00

(b)	0° inclination			3° inclination			90° inclination					
SH Treated	10 W	20 W	30 W	10 W	20 W	30 W	10 W	20 W	30 W			
0 %	40.15	63.25	72.59	40.15	63.25	72.59	40.15	63.25	72.59			
20 %	19.35	31.20	30.75	21.80	31.00	34.20	21.7	23.40	31.65			
40 %	19.13	25.80	23.40	17.90	25.80	27.30	18.9	19.80	24.30			
60 %	14.25	19.40	21.15	17.60	19.40	19.50	15.6	19.60	21.60			
80 %	19.7	19.20	28.50	13.40	19.10	18.45	15.8	21	23.10			
100 %	20.7	17.00	24.90	13.50	16.00	19.05	20.5	23.20	23.25			
	(b)											

Figure 4.12: ΔT color map for copper. (a) Untreated and (b) SH-treated.

even a small amount of working fluid is typically sufficient to shift the cells to blue indicating that the chemically-treated base plate outperforms its untreated counterpart in terms of heat transfer.

- 2. The greatest benefits to applying a chemical coating are realized when the heat pipe is inclined at the intermediate angle $\alpha = 3^{\circ}$.
- 3. There is a greater benefit to applying a chemical coating when the base plate



Figure 4.13: Experimental images taken from the optical heat pipe at steady state conditions for a heat input of 30 W, an inclination angle of $\alpha=90^{\circ}$ and various fill fractions. Copper base plate

is composed of copper vs. aluminum. Correspondingly, figure 4.14 (b) shows more cells have a blue or dark blue shade than does figure 4.14 (a). From the condensation chamber experiment, it is verified from visual footage that the mode of condensation of untreated copper is filmwise and SH-treated copper is dropwise. This difference in mode of condensation is not applicable to aluminum because the untreated aluminum also exhibits dropwise condensation. As a result, the difference in heat transfer performance is more prominent in copper than in aluminum case.

Now we turn our attention to a comparison of a different kind, which is not a

Aluminum		0° inclina	tion	3°	3° inclination)° inclina		
Aluminum	10 W	20 W	30 W	10 W	20 W	30 W	10 W	20 W	30 W	
0 %	5.4	7.3	7.7	5.4	7.3	7.7	5.4	7.3	7.7	
20 %	0.5	28.3	0.7	4.8	9.4	3.5	4.4	11.3	20.4	Blue Upward
40 %	0.6	12.8	3.1	3.1	8.6	6.3	25.0	2.4	3.7	
60 %	3.2	6.5	3.6	2.3	4.4	3.3	7.5	11.9	8.9	Red
80 %	5.4	2.7	7.4	0.8	1.3	3.5	0.0	2.4	8.0	Downward
100 %	4.9	3.9	2.4	2.2	4.9	2.7	6.4	0.7	3.4	Indicator (%
					(a)					27.5
	0° inclination 3° inclination 90° inclination									
Copper	10 W	20 W	30 W	10 W	20 W	30 W	10 W	20 W	30 W	7.5
0 %	11.1	15.8	14.2	11.1	15.8	14.2	11.1	15.8	14.2	2.5
20 %	9.4	9.3	8.9	2.1	9.6	18.0	5.9	19.6	4.5	7.5
40 %	2.4	11.6	6.0	8.7	10.7	27.2	6.4	16.1	25.0	
60 %	16.9	11.0	5.4	2.6	11.0	25.7	8.5	2.5	8.9	
80 %	9.4	11.9	15.3	38.4	12.8	25.0	23.7	1.9	10.5	
100 %	4.6	7.6	23.6	37.8	13.5	14.2	3.3	4.1	14.8	
					(b)					

Figure 4.14: Performance indicator color plot: blue represents SH-treated performing better and red color represents untreated performing better (a) Aluminum (b) Copper

comparison between untreated and SH-treated samples. Rather this is a comparison of the condensation pattern of figures 4.11 and 4.13 with that of figures 3.11(c) and 3.12(c). To highlight the differences in the condensation patterns, consider the image pairs reproduced in figures 4.15 (aluminum) and 4.16 (copper). In both cases, we visualize the condensation of vapor on untreated and SH-treated samples for both experimental set-ups. The pairs of images (condensation chamber vs. optical heat pipe) exhibit some qualitative similarities, but they are not exactly the same. In interpreting the associated differences, it is important to appreciate that the condensation pattern depends on a variety of factors. Obviously, this list includes surface characteristics, which we have kept constant in both the condensation chamber and optical heat pipe by ensuring the same chemical treatment regime and therefore surface contact angles. Also important, however, is the degree of subcooling which is not the same for the condensation chamber and optical heat pipe experiments [43]. More specifically, the degree of subcooling for the condensation chamber experiment ranges from 5°C to 19°C but for optical heat pipe experiments this range is between 21.60°C and 51.45°C. Jin et al [43] in their molecular dynamics (MD) simulation study investigated the dependency of the condensation mechanism on the surface wettability and subcooling temperature. Dropwise condensation can take place by one of two mechanisms, namely condensation from nucleation and condensation from film rupture [43]. Which of these mechanisms is more important depends on the degree of subcooling. When the subcooling temperature is small and the surface contact angle is large such that the surface is hydrophobic or superhydrophobic surface, dropwise condensation by nucleation prevails. This is the reason why we observe so many spherical droplets in condensation chamber images. On the other hand when both the subcooling temperature and the contact angle are large, dropwise condensation is a byproduct of film rupture. In this case, there exists a thin film of condensate already present on the condensing surface. The condensing fluid is therefore already present on the subcooled surface as a result of adsorption. As more and more molecules adsorb on the surface, a liquid film grows until the point of contraction and rupture whereafter droplets may arise in conjunction with the remnants of the film. Given this difference of condensation mechanism, we expect, and indeed observe, different condensation patterns in the aforementioned image pairs.



Figure 4.15: Comparison of experimental images taken from the condensation chamber and optical heat pipe at steady state conditions for a heat input of 30 W, an inclination angle of $\alpha=90^{\circ}$ and 60 % fill fraction. Aluminum



Figure 4.16: Comparison of experimental images taken from the condensation chamber and optical heat pipe at steady state conditions for a heat input of 30 W, an inclination angle of $\alpha=90^{\circ}$ and 60 % fill fraction. Copper

4.5 Conclusion

The purpose of performing the optical heat pipe experiments was to examine the effect of surface chemistry on the heat transfer performance of the heat pipe. The

outcome of the optical heat pipe experiments are summarized as follows:

- The optical heat pipe experiments identify those cases where having a superhydrophobic chemical coating in the condenser section is beneficial. In general, such benefits accrue when the angle of inclination is at intermediate angle $\alpha = 3^{\circ}$.
- The maximum performance difference between untreated and SH-treated base plates is found to be 28.3% for aluminum and 38.4% for copper.
- This study also concludes that the benefit of having a chemical coating in a copper heat pipe is greater than the benefit realized in an aluminum heat pipe.

Chapter 5

Concluding remarks and future recommendation

In this thesis, every chapter concludes with a recap of the main outcomes and pertinent discoveries, whether these stem from the current experiments or those conducted by others. Therefore, this final chapter aims to offer a broader view of the collective impact of all the results within the research context. Various industries are exploring the potential of surface chemical treatments to improve heat pipe performance. It is, vital, therefore, to determine the types of surface chemical treatments that offer not just enhanced heat transfer capabilities but also economic viability. A more effective approach involves examining the condensation rate, which directly correlates with the heat transfer performance of heat pipes.

5.1 Overview and summary

The experiments performed and discussed in Chapter 3 explore the condensation performance of samples with different surface energies. The findings suggest that increasing hydrophobicity through the introduction of micro-nano structures and reducing surface energy can indeed boost the rate of droplet shedding. The visualization experiment offered valuable insights into both the quantity of droplets and the average droplet radius. But the question still remains whether having such a chemical coating, that increases the flow of liquid from the heat pipe condenser section to the evaporator section can improve heat transfer performance or not. Indeed, a major drawback of the application of a surface chemical coating is they reduce the thermal conductivity of the base surface. Therefore, any increase in the rate of water shedding must more than compensate for the associated increase of conductive resistance. This particular concern is addressed in next set of experiments.

Chapter 4 discusses the experiments and results performed on a representative flat plate heat pipe termed as an "optical heat pipe" because it allows visual observations to be made of the condensation process. The main idea was to replicate the similar type of condensation behavior as observed in the condensation chamber experiment, in the condenser section of the heat pipe which can be further investigated with respect to ΔT (defined by equation 4.3) and the overall thermal resistance (defined by equation 4.4). The findings suggest that the impact of the surface chemical coating is most pronounced when the angle of inclination is small enough to minimize the distance that condensed droplets travel to return to the liquid pool that collects in the evaporator section. Also, copper gives a greater benefit of applying a surface coating when compared with aluminum. The results in some cases, which have been highlighted by blue regions in figure 4.14 aligned to our hypothesis that having increased droplet shedding can improve heat pipe performance, it is been observed that the droplet shape and condensate behavior has slight differences between the optical heat pipe footages vs. the condensation chamber footages. More specifically, the droplets that form inside the heat pipe show a greater irregularity of shape as compared to the more circular droplets visible within the condensation chamber. We believe the primary reason for this difference is related to the vapor flow rate. In the condensation chamber, the flow rate of humid air was lower than the optical heat pipe because the two experiments were designed to illustrate slightly different physics. We could not set the same flow rate of humid air for two sets of experiments because of experimental limitation. This has potentially changed the shape of droplets. However, the droplet shedding behavior is still the same for both the condensation chamber and optical heat pipe, which is the reason why we have a similar conclusion in both sets of experiments.

5.2 Future work

Surface chemical coatings enjoy a large application space in heat pipe technology because surface chemical coatings may be of different types based on the substrate material, working fluid and temperature regime. To apply the lessons learned from the experiments described above, other considerations have to be included. Accordingly, some of the scopes of future work are mentioned below:

- A significant consideration in the application of surface chemical coatings is their durability. Most HVAC manufacturers require heat pipes to operate for more than 20 years with minimal maintenance. Durability of the surface chemical coating discussed in this study has not been investigated which can be a future research topic if these kinds of coating are seriously considered for industrial deployment.
- To isolate the effects of surface chemistry, we have not considered grooved heat pipes, however, these might be worth exploring in future. In particular, it would be interesting to see if grooves coupled with a surface coating lead to the same fractional performance improvement seen here for the case of a heat pipe devoid of micro-structure.
- Our research focused primarily on the phenomena of dropwise condensation and the possibility of expediting liquid return to the evaporator. As such, application of a chemical coating was limited to the condenser section. However, it is also worth investigating the application of a hydrophobic (hydrophilic) surface coating in the adiabatic (evaporator) sections to determine what impact,

if any, such coatings may exert.

• In this study we characterized our surface based on surface contact angle rather than contact angle hysteresis. We did so because it is well understood that surfaces having higher contact angle will exhibit lower contact angle hysteresis which is favorable to droplet shedding process [78]. On the other hand, it still remains to study the precise relationship between droplet shedding and contact angle hysteresis more carefully at least in the context of condensation within the condenser section of a heat pipe which is what we are proposing as an idea for future studies.

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Appendix A: Surface chemical treatment

The details of the steps taken for the sample preparation are given in this appendix. The surface chemical treatment was performed using a two step process. The first step consists of etching and the second step consists lowering the surface energy [75, 76].

A.1 Chemicals

The chemicals used for the surface chemical treatment are listed in table A.1.

A.2 Material

The following materials were used for preparing the surface coating :

- Mirror-like multipurpose 110 copper sheeting was used for copper sample preparation and polished multipurpose 6061 aluminum was used for aluminum sample preparation both supplied by McMaster-Carr. The dimension of the samples were 25.4 mm × 25.4 mm × 3 mm (Length × Breadth × Thickness)
- Petri dishes
- Beakers
- Precision balance (Sartorius 1265 MP)
- Stirrer

Chemical list for surface coating preparation										
Chemicals	Supplier	Details								
Potassium hydroxide (KOH)	Fisher Scientific	Potassium hydrox- ide (pellets/certified ACS), 500 gm								
Ammonium persul- phate	MG Chemicals	410 Ammonium persulphate copper etchant, 1 Kg solid crystals								
Sodium hydroxide (NaOH)	Fisher Scientific	Sodium hydroxide (pellets/certified ACS), 1 kg								
Ethanol (C_2H_5OH)	LabChem	Reagent ethanol, ACS grade, 94.0% to 96.0% (v/v)								
Lauric acid	TCL America	Lauric acid $98.0{+}\%$								

Table A.1: Chemicals that were used in the preparation of the samples

A.3 Solution preparation

A.3.1 Etching solution preparation

The etching solution was different for the copper and aluminum samples. To prepare the aluminum sample etching solution, potassium hydroxide (KOH) and DI water were used; to prepare the copper sample etching solution, sodium hydroxide (NaOH), ammonium persulphate and DI water were used. Details concerning solution volumes and concentrations are given in tables A.2 and A.3.

A.3.2 Preparation of the solution to lower the surface energy

The solution to lower the surface energy of the samples was same for both aluminum and copper. Ethanol (C_2H_5OH) and lauric acid were used to prepare the solution. 2gm of lauric acid was mixed with 100 ml of ethanol to prepare the solution.

A.4 Surface chemical treatment steps

The following steps were taken to prepare the samples:

- Immerse the samples in etching solution to create micro and nano structures on the exposed surface.
- The etching time varies from sample to sample which is given in tables A.2 and A.3.
- Clean each extracted sample using ethanol and DI water.
- Immerse the samples for 60 minutes in lauric acid and ethanol solution to lower the surface energy.
- Clean the sample using ethanol and DI water and let it dry for 2 hours.

A schematic illustrating the above process is given in figure A.1.

Table A.2: Details of the etching time and etching solution: Aluminum

Sample	DI water (ml)	KOH (gm)	Concentration of etching solution (mo- larity)	Etching time (mins)	Contact angle (°)	Standard deviation (°)
Al S 1	100	1.00	0.18	30	117.97	3.21
Al S 2	100	1.00	0.18	45	136.59	0.81
Al S 3	100	8.40	1.50	40	154.35	1.31
AlS4	100	8.40	1.50	45	158.07	1.87



Figure A.1: Surface chemical treatment process for copper and aluminum samples (the second process step is the same for both aluminum and copper).

Table A.3: Details of the etching time and etching solution: Copper

Sample	DI water (ml)	NaOH (gm)	Ammonium persul- phate (gm)	Concentration of etching solution (mo- larity)	Etching time (mins)	Contact angle (°)	Standard deviation (°)
Cu S 1	100	10	2.28	2.50	30	144.17	2.98
Cu S 2	100	10	2.28	2.50	60	147.49	0.42
CuS3	100	10	2.28	2.50	90	152.79	0.48
CuS4	100	10	2.28	2.50	120	158.07	1.81
CuS 5	100	10	2.28	2.50	150	167.00	0.50
Appendix B: Image processing

B.1 Image Processing (Example)



Figure B.1: Example of image processing steps

Figure B.1 shows the resulting experimental image after each image processing step. The numbers in the frames represent the step order. The description of the frames shown in the figure B.1 are given as follows:

- 1. This was the original experimental image where ROI was defined.
- 2. The original footage was converted into monochrome to avoid complications due to the broad spectrum of color associated with manual thresholding step.

- 3. In this step, images were sharpened using the built-in sharpen tool of the software.
- 4. Images were then enhanced by manually adjusting the brightness, contrast and gamma values. This step was necessary to differentiate the droplets from the condensing surface.
- 5. Next, manual segmentation was performed using the software thresholding tool to identify the droplets based on light intensity. Figure B.2 shows a snapshot of the thresholding tool of the software.



Figure B.2: Thresholding tool of Image Pro Premier software.

6. After manual segmentation, the built-in count function was used to finally collect data from each of the droplets identified. The data in question consisted of droplet diameter and surface area.

Appendix C: Experimental data

C.1 Temperature data

In this appendix, the variation of temperature in the evaporator and condenser sections of the optical heat pipe is given. We consider aluminum and copper base plates and likewise instances where a surface chemical treatment has vs. has not been applied. The positions of the evaporator and condenser sections are given in figure C.1. Thermocouple data for the adiabatic section are not included because these data are not used in the overall thermal resistance calculations of chapter 4. Thermocouple readings are considered accurate to within $\pm 1^{\circ}$ C.



Figure C.1: Thermocouple position along the axial direction of the heat pipe T1 (0.30 cm), T2 (3.40), T3 (6.50 cm), T4 (9.60 cm), T5 (14.20 cm), T6 (22.20 cm), T7 (28.20 cm), T8 (36.20 cm), T9 (40.80 cm) and T10 (49.70 cm)

	Evaporator section				Condenser section	
Fill fraction (%)	T1	Τ2	Т3	T4	Т9	T10
0	48.15	48.41	48.34	48.09	7.84	7.78
20	42.15	42.40	42.30	42.05	9.54	9.60
40	40.98	40.58	40.31	40.82	9.41	9.53
60	40.85	40.75	40.87	40.93	9.59	9.41
80	41.03	41.05	40.88	40.85	9.37	9.12
100	41.67	41.55	41.48	41.55	9.63	9.49

Table C.1: Aluminum untreated: $\alpha=90^\circ$, $Q{=}~10\,\mathrm{W}$ different fill fraction

Table C.2: Aluminum untreated: $\alpha=90^\circ$, $Q{=}~20\,\mathrm{W}$ different fill fraction

	E	vaporat	Condenser section			
Fill fraction (%)	T1	Τ2	Т3	Τ4	Т9	T10
0	70.52	69.96	69.45	69.31	8.41	8.30
20	64.81	64.45	64.54	64.05	12.41	12.50
40	49.32	49.33	49.09	49.13	16.70	15.92
60	57.55	57.47	56.46	56.37	16.24	16.68
80	57.20	56.89	57.25	56.64	15.59	15.79
100	56.95	56.58	56.39	56.20	15.72	15.93

	Evaporator section				Condenser section	
Fill fraction $(\%)$	T1	Τ2	Т3	Τ4	Т9	T10
0	77.45	77.40	77.39	77.22	10.13	9.97
20	71.57	71.50	71.49	71.42	20.13	19.97
40	56.66	56.53	56.54	56.89	20.11	19.99
60	63.28	63.11	63.06	63.11	21.18	20.81
80	61.47	61.27	61.03	59.99	21.76	21.23
100	60.58	60.39	60.95	60.92	21.52	21.61

Table C.3: Aluminum untreated: $\alpha=90^\circ$, $Q{=}$ 30 W different fill fraction

Table C.4: Aluminum SH-treated: $\alpha=90^\circ$, $Q{=}~10\,\mathrm{W}$ different fill fraction

	E	vaporat	Condenser section			
Fill fraction (%)	T1	T2	Т3	Τ4	Т9	T10
0	48.07	48.06	47.95	48.00	5.70	5.13
20	44.17	44.16	44.05	44.10	12.70	13.13
40	35.99	35.89	35.88	35.90	12.81	12.21
60	42.22	42.35	42.15	42.03	13.34	13.02
80	45.13	45.11	44.84	44.81	12.95	13.59
100	46.35	46.64	46.60	46.38	12.27	12.63

	Evaporator section				Condenser section	
Fill fraction (%)	T1	Τ2	Т3	Τ4	Т9	T10
0	73.72	73.61	73.5	73.51	7.95	7.41
20	64.35	64.49	64.12	64.18	17.95	18.41
40	48.5	48.77	48.3	48.26	16.47	16.24
60	53.77	53.08	53.03	53.45	17.49	17.79
80	56.61	56.79	56.37	56.07	16.05	16.27
100	57.06	57.03	57.01	56.89	16.45	16.74

Table C.5: Aluminum SH-treated: $\alpha=90^\circ$, $Q{=}~20\,\mathrm{W}$ different fill fraction

Table C.6: Aluminum SH-treated: $\alpha=90^\circ$, $Q{=}~30\,\mathrm{W}$ different fill fraction

	E	vaporat	Condenser section			
Fill fraction (%)	T1	Τ2	Т3	Τ4	Т9	T10
0	80.76	81.08	81.01	81.41	8.13	8.97
20	61.26	61.58	61.51	62.01	20.45	20.84
40	58.45	58.39	58.68	58.49	20.23	20.89
60	60.20	60.15	58.61	58.05	20.73	20.96
80	59.81	58.94	57.88	55.88	21.91	21.75
100	59.45	59.05	58.75	58.79	21.02	21.41

	E	vaporat	Condenser section			
Fill fraction (%)	T1	Τ2	Т3	T4	Т9	T10
0	48.15	48.41	48.34	48.09	7.84	7.78
20	38.99	39.33	39.51	39.40	10.27	10.74
40	41.14	41.37	41.61	41.41	10.08	10.38
60	42.17	42.47	42.48	42.23	9.74	9.84
80	42.84	43.04	43.14	42.88	10.13	10.62
100	43.64	43.89	43.73	43.54	10.13	10.27

Table C.7: Aluminum untreated: $\alpha=0^\circ$, $Q{=}~10\,\mathrm{W}$ different fill fraction

Table C.8: Aluminum untreated: $\alpha=0^\circ$, $Q{=}~20\,\mathrm{W}$ different fill fraction

	E	vaporat	Condenser section			
Fill fraction (%)	T1	T2	Т3	Τ4	Т9	T10
0	70.52	69.96	69.45	69.31	8.41	8.30
20	63.09	63.52	63.64	63.19	15.34	15.98
40	59.36	59.86	60.08	60.14	16.98	16.75
60	49.22	49.6	49.94	49.84	17.39	17.51
80	58.95	59.42	59.81	59.68	15.80	15.13
100	69.29	69.56	68.36	67.44	14.92	15.02

	E	vaporat	Condenser section			
Fill fraction $(\%)$	T1	Τ2	Т3	Τ4	Т9	T10
0	77.45	77.40	77.39	77.22	10.13	9.97
20	60.48	60.90	61.23	60.78	20.77	20.83
40	58.70	59.79	59.20	58.85	20.40	20.18
60	62.31	62.86	63.28	63.27	20.98	20.88
80	65.23	65.93	66.45	66.30	19.69	19.27
100	69.98	69.84	69.61	68.92	20.16	20.02

Table C.9: Aluminum untreated: $\alpha=0^\circ$, $Q{=}~30\,\mathrm{W}$ different fill fraction

Table C.10: Aluminum SH-treated: $\alpha=0^\circ$, $Q{=}~10\,\mathrm{W}$ different fill fraction

	E	lvaporat	Condenser section			
Fill fraction (%)	T1	T2	Т3	Τ4	Т9	T10
0	48.07	48.06	47.95	48.00	5.70	5.13
20	41.65	41.59	41.51	41.14	12.63	12.42
40	43.89	43.84	43.44	43.35	12.52	12.04
60	44.52	44.51	44.26	44.09	12.94	12.75
80	43.64	43.37	43.16	43.01	12.23	12.66
100	44.78	44.57	44.35	44.21	12.95	12.31

	Evaporator section				Condenser section	
Fill fraction (%)	T1	Τ2	Т3	T4	Т9	T10
0	73.72	73.61	73.50	73.51	7.95	7.41
20	53.02	52.62	52.20	52.10	18.30	18.27
40	56.58	56.23	56.07	55.79	18.45	18.89
60	53.77	53.16	52.98	52.86	18.91	18.88
80	61.56	61.27	61.16	60.66	18.11	18.63
100	69.73	69.34	69.07	68.88	17.51	17.80

Table C.11: Aluminum SH-treated: $\alpha=0^\circ$, $Q{=}~20\,\mathrm{W}$ different fill fraction

Table C.12: Aluminum SH-treated: $\alpha=0^\circ$, $Q{=}~30\,\mathrm{W}$ different fill fraction

	E	vaporat	Condenser section			
Fill fraction (%)	T1	T2	Т3	Τ4	Т9	T10
0	80.76	81.08	81.01	81.41	8.13	8.97
20	60.30	60.28	60.25	59.36	20.55	20.05
40	60.68	60.33	60.26	60.01	20.09	20.45
60	64.29	64.01	63.79	63.22	20.61	20.03
80	71.71	71.76	71.79	70.83	21.33	21.83
100	69.10	69.07	68.58	67.72	20.28	20.37

	E	vaporat	Condenser section			
Fill fraction (%)	T1	Τ2	Т3	T4	Т9	T10
0	48.15	48.41	48.34	48.09	7.84	7.78
20	43.22	43.56	43.74	43.63	11.24	11.54
40	42.32	42.55	42.79	42.59	11.09	11.83
60	42.12	42.42	42.43	42.18	11.76	11.81
80	41.23	41.43	41.53	41.27	11.18	11.64
100	41.56	41.81	41.65	41.46	11.57	11.37

Table C.13: Aluminum untreated: $\alpha=3^\circ$, $Q{=}~10\,\mathrm{W}$ different fill fraction

Table C.14: Aluminum untreated: $\alpha=3^\circ$, $Q{=}~20\,\mathrm{W}$ different fill fraction

	E	lvaporat	Condenser section			
Fill fraction (%)	T1	T2	Т3	Τ4	Т9	T10
0	70.52	69.96	69.45	69.31	8.41	8.30
20	65.78	66.21	66.33	65.88	16.35	16.14
40	49.00	49.50	49.72	49.78	17.02	16.98
60	61.84	62.22	62.56	62.46	17.35	16.99
80	64.02	64.49	64.88	64.75	17.21	17.25
100	67.40	67.67	66.47	65.55	17.33	17.61

	Evaporator section				Condenser section	
Fill fraction $(\%)$	T1	Τ2	Т3	T4	Т9	T10
0	77.45	77.40	77.39	77.22	10.13	9.97
20	68.29	68.71	69.04	68.59	21.54	21.87
40	54.38	55.47	54.88	54.53	21.56	21.16
60	61.84	62.39	62.81	62.80	21.64	21.98
80	63.37	64.07	64.59	64.44	21.66	21.97
100	65.52	65.38	65.15	64.46	21.16	21.49

Table C.15: Aluminum untreated: $\alpha=3^\circ$, $Q{=}$ 30 W different fill fraction

Table C.16: Aluminum SH-treated: $\alpha=3^\circ$, $Q{=}~10\,\mathrm{W}$ different fill fraction

	E	lvaporat	Condenser section			
Fill fraction (%)	T1	T2	Т3	Τ4	Т9	T10
0	48.07	48.06	47.95	48.00	5.70	5.13
20	46.31	46.25	46.17	45.80	15.64	15.42
40	46.06	46.01	45.61	45.52	15.64	15.66
60	45.94	45.93	45.68	45.51	15.99	15.94
80	45.88	45.61	45.40	45.25	15.67	15.99
100	45.92	45.71	45.49	45.35	16.01	16.22

	E	vaporat	Condenser section			
Fill fraction $(\%)$	T1	Τ2	Т3	T4	Т9	T10
0	73.72	73.61	73.50	73.51	7.95	7.41
20	66.30	65.90	65.48	65.38	20.45	20.87
40	50.84	50.49	50.33	50.05	20.97	20.48
60	64.24	63.63	63.45	63.33	20.97	20.16
80	67.81	67.52	67.41	66.91	20.87	20.55
100	68.73	68.34	68.07	67.88	21.02	21.69

Table C.17: Aluminum SH-treated: $\alpha=3^\circ$, $Q{=}~20\,\mathrm{W}$ different fill fraction

Table C.18: Aluminum SH-treated: $\alpha=3^\circ$, $Q{=}~30\,\mathrm{W}$ different fill fraction

	E	vaporat	Condenser section			
Fill fraction (%)	T1	T2	Т3	Т4	Т9	T10
0	80.76	81.08	81.01	81.41	8.13	8.97
20	68.26	68.24	68.21	67.32	22.45	22.97
40	54.26	53.91	53.84	53.59	22.46	22.64
60	62.42	62.14	61.92	62.35	22.34	22.97
80	63.72	63.77	63.80	62.84	22.98	22.49
100	65.34	65.31	64.82	63.96	22.35	22.16

	Evaporator section				Condenser section	
Fill fraction $(\%)$	T1	Τ2	Т3	Τ4	Т9	T10
0	44.78	44.53	44.35	44.31	8.45	8.24
20	30.74	31.17	30.92	30.70	10.49	10.38
40	30.25	30.60	30.36	29.97	10.20	9.99
60	29.04	28.47	28.36	27.73	11.19	11.52
80	30.52	30.24	30.02	29.73	9.55	9.31
100	31.09	30.72	30.60	30.45	9.35	9.68

Table C.19: Copper untreated: $\alpha=90^\circ$, $Q{=}~10\,\mathrm{W}$ different fill fraction

Table C.20: Copper untreated: $\alpha=90^\circ$, $Q{=}~20\,\mathrm{W}$ different fill fraction

	E	vaporat	Condenser section			
Fill fraction (%)	T1	Τ2	Т3	Τ4	Т9	T10
0	65.53	64.99	64.59	64.71	10.45	10.24
20	45.58	45.04	44.64	44.76	15.89	15.92
40	40.37	39.54	39.47	38.99	16.04	15.95
60	37.35	36.54	36.21	35.47	16.25	16.34
80	38.95	38.15	37.86	37.36	16.59	16.78
100	41.44	40.59	40.39	39.98	16.61	16.20

	E	vaporat	Condenser section			
Fill fraction $(\%)$	T1	Τ2	Т3	T4	Т9	T10
0	77.48	76.77	76.35	76.36	13.15	13.24
20	52.58	51.87	51.45	51.46	18.70	18.68
40	51.14	50.17	50.01	49.69	17.96	17.75
60	43.63	42.72	42.18	41.49	18.89	18.73
80	45.15	44.20	43.79	43.37	18.59	18.08
100	47.29	46.11	45.88	45.25	18.97	18.70

Table C.21: Copper untreated: $\alpha=90^\circ$, $Q{=}~30\,\mathrm{W}$ different fill fraction

Table C.22: Copper SH-treated: $\alpha=90^\circ$, $Q{=}~10\,\mathrm{W}$ different fill fraction

	E	vaporat	Condenser section			
Fill fraction (%)	T1	T2	Т3	Τ4	Т9	T10
0	47.81	47.46	47.31	47.54	7.25	7.51
20	32.91	32.56	32.41	32.64	11.05	10.81
40	29.85	29.73	29.66	29.35	10.88	10.62
60	28.15	27.82	27.46	26.83	12.08	11.86
80	26.63	26.02	25.85	25.12	10.14	10.08
100	33.34	32.96	32.51	32.33	12.33	12.25

	E	vaporat	Condenser section			
Fill fraction $(\%)$	T1	T2	Т3	T4	Т9	T10
0	71.98	71.55	71.37	71.55	8.24	8.49
20	42.12	41.69	41.51	41.69	18.66	18.06
40	39.26	38.97	38.81	38.79	19.14	19.18
60	40.10	39.83	39.21	39.42	20.05	20.03
80	39.64	39.49	39.27	39.36	18.41	18.48
100	41.94	41.60	41.25	41.74	18.29	18.59

Table C.23: Copper SH-treated: $\alpha=90^\circ$, $Q{=}~20\,\mathrm{W}$ different fill fraction

Table C.24: Copper SH-treated: $\alpha=90^\circ$, $Q{=}~30\,\mathrm{W}$ different fill fraction

	E	vaporat	Condenser section			
Fill fraction (%)	T1	T2	Т3	Τ4	Т9	T10
0	84.02	83.31	82.90	82.89	10.70	10.68
20	51.08	50.37	49.96	49.95	18.70	18.68
40	43.12	42.98	42.03	41.35	18.14	18.50
60	40.07	39.85	39.99	39.96	18.24	18.50
80	42.02	41.94	41.94	41.24	18.59	18.79
100	43.59	43.42	42.27	41.13	19.50	19.51

	Evaporator section				Condenser section	
Fill fraction $(\%)$	T1	T2	Т3	Τ4	Т9	T10
0	44.78	44.53	44.35	44.31	8.45	8.24
20	32.75	32.51	32.30	32.08	11.08	11.05
40	31.09	30.68	30.66	30.34	11.16	11.04
60	28.98	28.68	28.29	28.45	11.38	11.49
80	34.03	33.75	33.49	33.27	11.86	11.92
100	33.90	33.62	33.36	33.37	11.91	11.82

Table C.25: Copper untreated: $\alpha=0^\circ$, $Q{=}~10\,\mathrm{W}$ different fill fraction

Table C.26: Copper untreated: $\alpha=0^\circ$, $Q{=}~20\,\mathrm{W}$ different fill fraction

	Evaporator section				Condenser section	
Fill fraction (%)	T1	T2	Т3	Τ4	Т9	T10
0	65.53	64.99	64.59	64.71	10.45	10.24
20	49.07	48.65	48.16	48.07	14.10	14.09
40	44.18	43.70	43.24	43.03	14.27	14.41
60	37.24	36.83	36.53	35.88	14.77	14.87
80	35.63	35.29	34.51	34.84	13.42	13.13
100	32.29	32.20	31.77	32.23	13.75	13.70

	Evaporator section				Condenser section	
Fill fraction (%)	T1	Τ2	Т3	Τ4	Т9	T10
0	77.48	76.77	76.35	76.36	13.15	13.24
20	52.54	51.92	51.49	51.13	18.11	17.94
40	43.92	43.23	42.37	42.80	18.17	18.19
60	40.77	40.12	39.68	39.02	17.55	17.56
80	49.53	49.05	48.44	48.32	18.04	17.83
100	45.61	44.83	44.49	43.80	18.17	18.40

Table C.27: Copper untreated: $\alpha=0^\circ$, $Q{=}$ 30 W different fill fraction

Table C.28: Copper SH-treated: $\alpha=0^\circ$, $Q{=}~10\,\mathrm{W}$ different fill fraction

	Evaporator section				Condenser section	
Fill fraction (%)	T1	T2	Т3	Τ4	Т9	T10
0	47.81	47.46	47.31	47.54	7.25	7.51
20	34.60	33.94	33.74	33.55	14.43	14.80
40	34.38	34.37	34.14	34.35	15.00	15.36
60	31.11	30.71	30.58	30.33	16.38	16.49
80	34.11	34.27	33.60	33.72	14.20	14.26
100	34.64	34.19	34.02	33.57	13.19	13.63

	Evaporator section				Condenser section	
Fill fraction $(\%)$	T1	T2	Т3	T4	Т9	T10
0	71.98	71.55	71.37	71.55	8.24	8.49
20	48.33	47.78	47.60	47.38	16.75	16.40
40	44.64	43.83	43.49	42.84	17.95	19.86
60	37.10	37.03	36.81	37.14	17.67	17.58
80	38.20	37.04	36.62	35.67	17.98	17.39
100	43.95	43.91	43.23	43.16	18.48	18.85

Table C.29: Copper SH-treated: $\alpha=0^\circ$, $Q{=}~20\,\mathrm{W}$ different fill fraction

Table C.30: Copper SH-treated: $\alpha=0^\circ$, $Q{=}$ 30 W different fill fraction

	Evaporator section				Condenser section	
Fill fraction (%)	T1	T2	Т3	Τ4	Т9	T10
0	84.02	83.31	82.90	82.89	10.70	10.68
20	41.59	40.97	40.54	40.18	19.11	18.94
40	42.95	42.48	42.36	42.52	19.17	19.19
60	40.66	40.31	40.06	39.91	19.02	19.16
80	48.80	47.61	47.30	46.07	19.02	18.87
100	44.76	44.64	44.29	44.26	19.77	19.41

	Evaporator section				Condenser section	
Fill fraction (%)	T1	Τ2	Т3	Τ4	Т9	T10
0	44.78	44.53	44.35	44.31	8.45	8.24
20	37.18	36.98	35.80	35.05	13.20	13.31
40	35.67	35.41	35.28	35.03	13.12	12.98
60	34.24	34.16	33.95	33.76	13.16	13.31
80	29.53	29.39	29.07	29.02	14.36	14.16
100	30.41	30.18	29.92	29.76	14.77	15.17

Table C.31: Copper untreated: $\alpha=3^\circ$, $Q{=}~10\,\mathrm{W}$ different fill fraction

Table C.32: Copper untreated: $\alpha=3^\circ$, $Q{=}~20\,\mathrm{W}$ different fill fraction

	E	vaporat	Condenser section			
Fill fraction (%)	T1	T2	Т3	Τ4	Т9	T10
0	65.53	64.99	64.59	64.71	10.45	10.24
20	52.58	52.33	50.33	48.86	16.55	16.91
40	45.43	45.06	44.64	44.28	16.08	15.84
60	38.81	38.60	37.93	37.83	16.47	16.53
80	39.45	39.18	38.54	38.46	17.18	16.84
100	37.69	37.38	36.68	36.64	18.80	18.40

	Evaporator section				Condenser section	
Fill fraction (%)	T1	Τ2	Т3	Τ4	Т9	T10
0	77.48	76.77	76.35	76.36	13.15	13.24
20	62.95	63.08	60.06	58.27	19.26	19.52
40	57.39	57.09	56.45	56.41	19.46	19.22
60	46.02	45.67	44.98	44.60	19.23	18.91
80	44.97	44.55	43.77	43.74	19.69	19.63
100	42.72	42.23	41.55	41.27	19.93	19.56

Table C.33: Copper untreated: $\alpha=3^\circ$, $Q{=}$ 30 W different fill fraction

Table C.34: Copper SH-treated: $\alpha=3^\circ$, $Q{=}~10\,\mathrm{W}$ different fill fraction

	Evaporator section				Condenser section	
Fill fraction (%)	T1	Τ2	Т3	Τ4	Т9	T10
0	47.81	47.46	47.31	47.54	7.25	7.51
20	39.23	39.03	37.84	36.10	16.20	16.31
40	34.74	34.31	34.18	33.73	16.07	16.61
60	34.51	33.85	33.72	33.12	16.11	16.30
80	31.29	30.84	30.35	30.76	17.58	17.25
100	31.58	30.77	30.77	30.94	17.39	17.65

	Evaporator section				Condenser section	
Fill fraction $(\%)$	T1	T2	Т3	T4	Т9	T10
0	71.98	71.55	71.37	71.55	8.24	8.49
20	49.96	49.98	50.43	50.58	19.21	19.28
40	45.46	45.90	45.54	45.36	19.99	19.55
60	40.28	39.54	39.46	38.65	20.13	20.04
80	37.57	36.82	36.75	35.93	17.91	17.44
100	37.93	37.83	37.07	37.67	21.52	21.74

Table C.35: Copper SH-treated: $\alpha=3^\circ$, $Q{=}~20\,\mathrm{W}$ different fill fraction

Table C.36: Copper SH-treated: $\alpha=3^\circ$, $Q{=}$ 30 W different fill fraction

	E	vaporat	Condenser section			
Fill fraction (%)	T1	Τ2	Т3	Τ4	Т9	T10
0	84.02	83.31	82.90	82.89	10.7	10.68
20	58.12	57.78	57.09	56.98	23.23	23.36
40	51.27	51.23	50.38	50.37	23.33	23.70
60	41.57	40.73	40.65	39.69	21.17	21.16
80	40.70	39.77	39.76	38.70	21.39	21.69
100	41.89	40.62	40.55	39.28	21.39	21.69

Appendix D: CAD drawings

The following computer-aided-designs were drawn using SolidWorks Educational version from Dassault Systems SolidWorks Corp. The designs in pages 115 to 117 represent different layers of the optical heat pipe. Those in pages 119 to 121 represent thermocouple adapters which are used to allow thermosouples to take temperature measurements from the evaporator and the adiabatic sections of the optical heat pipe. The design in page 121 represents an extension to hold the optical heat pipe in place when bolted with the tilting table.















